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DESIGN, SELECTION, AND INSTALLATION OF AIRCRAFT HEAT EXCHANGERS

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FOREWORD

At the present time the aircraft industry is undergoing rapid expansion and a relatively large number of new men are working in the aircraft-cooling field. The amount of training that many of these men have had in the general field of heat transfer and particularly in the specialized field of heat transfer in connection with aircraft is necessarily rather limited. Some of them are consequently confronted with the problem not only of becoming oriented in the field of heat transfer but also of trying to read, assimilate, and evaluate in a limited time the large number of reports that have been published on the design, selection, and installation of aircraft heat exchangers.

Realizing that the above conditions existed, the Bureau of Aeronautics requested the National Advisory Committee for Aeronautics to prepare a report summarizing present knowledge regarding heat exchangers as applied to aircraft.

The authors of the present report believe that the problem can be alleviated by use, as a textbook, of a report in which the present knowledge about the elements of design, selection, and installation is summarized. The authors also believe that men experienced in the field can use as a reference book a report in which the pertinent data, which have been published in many separate papers, are collected. They hope that the present report can fill both these needs; that it will be found useful both as a textbook and as a reference book.

It is not intended that this report entirely take the place of standard texts on heat transfer, which give thorough treatments of heat transfer in general. It is neither necessary nor desirable to include in the present report a detailed treatment of the mechanism of fluid flow and of heat transfer. For such a treatment, including a discussion of turbulent boundary layers, analogies between heat transfer and fluid friction, and so forth, reference 1 is recommended.

Inasmuch as a great deal of research is being done on the subjects covered by the present report, revision will probably be desirable from time to time. The authors will welcome comments and suggestions for improvements in presentation and scope of the material included in the report.

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SUMMARY

A survey of the subject of aircraft heat exchangers is given in the present report. The report is divided into three main parts. Part I, entitled "Design," presents the fundamental relations for calculating heat-transfer rate and pressure loss in heat exchangers, numerical data for use in applying these relations to various designs, energy-balance equations for fluid flow, methods of calculating the power cost of heat exchangers, and a discussion of the application of the foregoing material to the design of heat exchangers. Part II, entitled "Selection," discusses the selection of the external dimensions of coolant radiators, air intercoolers, oil coolers, and engine fins. Part III, entitled "Installation," considers the problems of installing heat exchangers in aircraft; namely, the design of engine cowlings, wing entrances, scoops, exits, and ducts. Appendixes give the physical properties of air, the properties of standard atmosphere as defined by the Navy, the Army, and the NACA; and an impact-pressure chart.

INTRODUCTION

A short time ago the power produced by the power plants of airplanes was small and the altitude of operation of airplanes was low. On account of the small engine power, the inefficiency with which that power was developed, and the fact that long range was not required of airplanes, the efficiency with which cooling was obtained was a relatively unimportant consideration. On account of the low altitudes of operation, the cooling problem was not a difficult one. Heat exchangers were usually selected to have as small a volume as possible and still be able to function on the available pressure drop.

Efforts are being directed currently toward increasing the range of bombers and cargo airplanes. The need for greater range and carrying capacity and the large engine powers that are used today make it important that cooling be obtained efficiently. The high altitudes at which present-day airplanes operate also make the cooling problem more difficult than it formerly was.

The efficiency and the cooling capacity of heat exchangers generally can be increased by: (1) improving the design of the elements of which they are constructed, (2) selecting better proportions for their external dimensions, and (3) improving their installation in the airplane. The present report is therefore divided into three main parts: I - Design; II - Selection; and III - Installation. Part I is an attempt to treat in simple terms the fundamental relations by which heat-transfer rate, pressure drop, and power consumption in heat exchangers are calculated. It also contains a résumé of the experimental data on the heat-transfer and pressure-drop characteristics of many designs. Part II treats methods of selecting the external dimensions of heat exchangers of a given design. Part III is a consideration of the problems of design of the air entries, ducts, and exits that are associated with heat exchangers.

SYMBOLS

$a = \sqrt{\frac{2h}{k_m t}}$, 1/feet

A	cross-sectional area of fluid passage, square feet open frontal area of heat exchanger, square feet
A_s	projected frontal area of scoop, square feet
c	speed of sound, feet per second
c	guide-vane chord, feet
c_p	specific heat of fluid at constant pressure, Btu per pound per $^{\circ}$F
c_v	specific heat of fluid at constant volume, Btu per pound per $^{\circ}$F
C	constant in definitions of generalized radiator variables, foot-pounds per second per cubic foot of open radiator volume

C_D/C_L	ratio of drag coefficient to lift coefficient of airplane
C_{D_s}	drag coefficient of scoop, based on projected frontal area
C_1, C_2, C_3, C_4	numerical constants
D	hydraulic diameter of passage, feet diameter or width of duct, feet
D	drag, pounds
f	fin effectiveness, dimensionless friction factor, dimensionless ratio of cross-sectional areas of passages, dimensionless
f_R	ratio of open to total frontal area of radiator
F	mechanical energy lost per unit weight of fluid because of friction, foot-pounds per pound
g	acceleration due to gravity, feet per second per second
h	coefficient of heat transfer, Btu per second per square foot per $^{\circ}F$
h_a	coefficient of heat transfer based on $\overline{\Delta T_a}$, Btu per second per square foot per $^{\circ}F$
h_i	coefficient of heat transfer based on $\overline{\Delta T_i}$, Btu per second per square foot per $^{\circ}F$
h_S	local coefficient of heat transfer at dS, Btu per second per square foot per $^{\circ}F$
h_t	over-all coefficient of heat transfer between two fluids, Btu per second per square foot per $^{\circ}F$
h_x	local coefficient of heat transfer at x, Btu per second per square foot per $^{\circ}F$
H	rate of heat transfer, Btu per second
H	total pressure, pounds per square foot
ΔH	loss in total pressure, pounds per square foot

J	mechanical equivalent of heat (778 foot-pounds per Btu)
k	loss coefficient for duct, diffuser, corner, and so forth ($\Delta H/q$)
k	thermal conductivity of fluid, Btu per second per square foot per $^{\circ}F$ per foot
k_m	thermal conductivity of metal, Btu per second per square foot per $^{\circ}F$ per foot
k_p	pressure coefficient $\left(\frac{p_3}{p_0}\right)$
K_1, K_2	constants in definitions of generalized radiator variables
l	exponent in equation (5)
L	length of fluid passage, feet
L_c	length of cooling-air passage, feet
L_e	length of engine-air passage, feet
L_n	length of intercooler in no-flow direction, feet
m	exponent in equation (5)
m_n	center-to-center tube spacing normal to direction of air flow, feet
m_p	center-to-center tube spacing parallel to direction of air flow, feet
M	Mach number
n	number of tubes
p	static pressure, pounds per square foot
$\frac{p_{alt}}{p_{50,000}}$	ratio of pressure rise across blower at altitude to pressure rise across blower at 50,000 feet
Δp_c	static-pressure drop of cooling air, pounds per square foot
Δp_f	static-pressure drop due to friction, pounds per square foot

Δp_m	static-pressure drop due to momentum increase, pounds per square foot
P	net power cost of cooling, foot-pounds per second
P_c	power for pumping cooling air, foot-pounds per second
P_s	power for overcoming scoop drag, foot-pounds per second
P_t	total power cost of heat exchanger, foot-pounds per second
P_W	power cost of transporting weight of heat exchanger, foot-pounds per second
$\dfrac{P_{alt}}{P_{50,000}}$	ratio of power cost of blower at altitude to power cost of blower at 50,000 feet
q	dynamic pressure, pounds per square foot
Q	volume rate of flow, cubic feet per second
Q_c	volume rate of flow of cooling air, cubic feet per second
Q_e	volume rate of flow of engine air, cubic feet per second
r_b	radius from center of cylinder to root of fin, feet
R	Reynolds number, dimensionless ($R = \rho VD/\mu$)
R	gas constant in $pv = RT$
R	radius of curvature of duct, feet
R_d	radius of curvature at diffuser exit, feet
R_u	radius of curvature at diffuser entrance, feet
s	fin spacing, feet
s	guide-vane spacing or gap, feet
S	surface area for heat transfer, square feet

S_b	area of cylinder wall, square feet
S_d	area of unfinned part of heat-transfer surface, square feet
S_f	area of fin surface, square feet
S_t	surface area for heat transfer between two fluids, square feet
t	fin thickness, feet wall thickness, feet
T	temperature, $^{\circ}F$
T_c	temperature of cold fluid, $^{\circ}F$
T_f	temperature of fluid, $^{\circ}F$
T_h	temperature of hot fluid, $^{\circ}F$
T_w	temperature of wall, $^{\circ}F$
ΔT	temperature difference, $^{\circ}F$
$\overline{\Delta T}$	mean temperature difference; logarithmic-mean temperature difference, $^{\circ}F$
$\overline{\Delta T_a}$	arithmetic-mean temperature difference, $^{\circ}F$
ΔT_c	rise in temperature of cold fluid, $^{\circ}F$
ΔT_f	temperature difference between fin and fluid, $^{\circ}F$
ΔT_d	temperature difference between direct surface and fluid, $^{\circ}F$
ΔT_h	drop in temperature of hot fluid, $^{\circ}F$
U	internal or intrinsic energy per unit weight of fluid, Btu per pound
v	volume of heat exchanger, cubic feet open volume of heat exchanger, cubic feet
v	specific volume of gas, cubic feet per pound $(1/g\rho)$
V	speed of fluid, feet per second

V_{max} speed of air at minimum cross section of tube bank, feet per second

w fin width, feet

W weight rate of flow of fluid, pounds per second

W_c weight rate of flow of cooling air, pounds per second

W_e weight rate of flow of engine air, pounds per second

W_h weight rate of flow of hot fluid, pounds per second

W_m total mechanical work done by unit weight of fluid, foot-pounds per pound

W rate at which mechanical energy is recoverable from heat input, foot-pounds per second

W duct height, feet

W_e net external mechanical work done on unit weight of fluid, foot-pounds per pound

x distance from tube entrance of cross section at which h_x is measured, feet

γ ratio of specific heats (c_p/c_v)

ϵ factor to account for weight of heat-exchanger mounting, dimensionless

ζ mean temperature difference between hot fluid and cold fluid divided by inlet temperature difference, dimensionless

η rise in temperature of cold fluid divided by inlet temperature difference, dimensionless

η efficiency of diffuser, dimensionless

η efficiency of supercharger, dimensionless

θ one-half the included angle of expansion of diffuser, degrees

θ angle through which air is deflected by duct corner, degrees

μ coefficient of viscosity of fluid, slugs per foot-second

ξ drop in temperature of hot fluid divided by inlet temperature difference, dimensionless

ρ mass density of fluid, slugs per cubic foot

ρ_R density of radiator based on open volume, pounds per cubic foot

σ ratio of density at altitude to density at sea level, dimensionless

Φ dimensionless group for correlating laminar-flow heat-transfer coefficients, figure 5 $\left(\text{Prandtl number} \times \text{Reynolds number} \times \frac{D}{L}\right)$

Subscripts:

1, 2 two fluids in a heat exchanger

0, 1, 2, 3 stations shown in figure 17

i inlet

o outlet

A superscript bar indicates a mean value, and a prime indicates a generalized variable. Symbols in equations and figures taken from references have been changed throughout to the notation of the present paper.

I - DESIGN

Every design of heat exchanger can be visualized as an arrangement of tubes. Conventional coolant radiators and oil coolers are composed of bundles of circular tubes through which the cooling air flows and over which the liquid flows. Some intercoolers are likewise composed of bundles of circular tubes. Others are made up of flat plates as dividing surfaces between the two fluids, with indirect cooling surfaces attached. The air flows through passages that may be considered tubes. The passages that are formed by the cylinder walls, the fins, and the baffles of an air-cooled engine may likewise be thought of as tubes.

In the design of a heat exchanger, it is desirable that the heat-transfer and the pressure-drop characteristics of the elements or tubes be known. Part I is concerned with the factors that influence these characteristics; namely, the dimensions, the shape, the spacing, and the arrangement of the tubes, and the velocity and the physical properties of the fluid.

The flow of fluids through the tubes or passages can in general be characterized as laminar or turbulent. In laminar flow, each particle of the fluid flows in a line that is nearly parallel with the axis of the passage. There is practically no mixing of various parts of the fluid. In turbulent flow, the particles of the fluid move in an agitated, disorderly, eddying type of motion. Rate of heat transfer and frictional pressure drop are different for the two types of flow. The type of flow that takes place in a tube depends principally on the Reynolds number. The Reynolds number for flow through a tube is defined as the dimensionless group $\rho VD/\mu$ where D is to be taken as the hydraulic diameter of the tube. The critical Reynolds number below which laminar flow always occurs and above which turbulent flow usually occurs is about 2100 for a circular tube, provided that the length-diameter ratio of the tube is large enough for turbulent flow to be developed. For the tubes of small length-diameter ratio commonly used in aircraft heat exchangers, the flow may be considered turbulent if the Reynolds number is not less than 6000 to 8000.

In correlating data on fluid flow, the concept of hydraulic diameter is useful. The hydraulic diameter of a tube of any cross-sectional shape is defined as four times the cross-sectional area of the tube divided by the perimeter of the tube. For example, the hydraulic diameter of a

rectangular tube of sides a and b is $2ab/(a + b)$. The hydraulic diameter of a circular tube is the geometric diameter of the tube.

HEAT TRANSFER

In a study of the transfer of heat in aircraft heat exchangers, interest is centered on transfer by forced convection. The role played by radiation is a relatively unimportant one. Although conduction through solids is a necessary factor in the functioning of heat exchangers, a detailed consideration of conduction is not necessary. The treatment of heat transfer in the present report is therefore primarily a treatment of heat transfer by forced convection.

A foundation for the discussion of the heat transfer between the two fluids of a heat exchanger is laid by a discussion of the heat transfer between a fluid and a surface.

HEAT TRANSFER - BETWEEN FLUID AND SURFACE

Two Fundamental Equations

Consider a cold fluid flowing past a hot surface. The temperature difference that exists between the wall and the main body of the fluid is shown in figure 1. Experiment

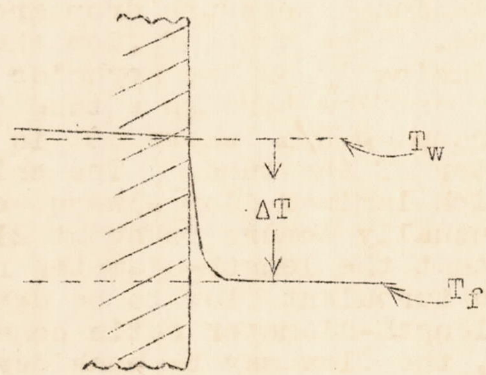

Figure 1. - Temperature difference between hot wall and cold fluid.

shows that dH, the amount of heat transferred per unit time to the fluid from an element of surface area dS, is

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proportional to the area of the surface element and to the temperature difference ΔT existing between the surface and the body of the fluid at the element dS; that is,

$$dH \propto \Delta T \, dS$$

If a proportionality factor h_s is used, the equation relating these quantities is

$$dH = h_s \, \Delta T \, dS \tag{1}$$

The rate of heat transfer from the entire surface S to the fluid is given by integration of equation (1). In the general case, h_s is a function of the temperature of the fluid, which varies over the surface and is a function of H. For air, however, the value of h_s is practically independent of temperature and h_s may be replaced by h. The temperature difference ΔT is a function of both the fluid and the surface temperatures, which are functions of H. The integration of equation (1) may therefore be indicated as

$$\int \frac{dH}{\Delta T} = h \int dS \tag{2}$$

If the integration of equation (2) is performed,

$$H = hS \, \overline{\Delta T} \tag{3}$$

where $\overline{\Delta T}$ is the mean temperature difference between the entire surface and the fluid. The actual integration shows that $\overline{\Delta T}$ is the logarithmic-mean temperature difference, defined as

$$\overline{\Delta T} = \frac{\Delta T_i - \Delta T_o}{\log_e \dfrac{\Delta T_i}{\Delta T_o}}$$

Equation (3) is the fundamental equation on which is based the treatment of heat transfer between a fluid and a surface by forced convection.

The transfer of heat, which is determined by equation (3), must also satisfy the energy-balance equation

$$H = W \, c_p \, (T_o - T_i) \tag{4}$$

(A more precise form of the energy-balance equation, which must be used occasionally, is given later in equation (40).) Calculations for the prediction of the heat-transfer rate in

a given piece of apparatus must take into account both equations (3) and (4). An example of the combination of these equations into a single working equation is given later in the discussion of the application of heat transfer to radiators.

Coefficient of Heat Transfer h

The quantity h appearing in equation (3) is known as the coefficient of heat transfer. Experiments show that the value of h depends on the type of flow, the arrangement of the surface area, the hydraulic diameter of the passage, and the velocity and the physical properties of the fluid. It is the purpose of the present section to give equations and data that will permit the calculation of the value of h for the various conditions encountered in aircraft heat exchangers.

Turbulent flow through straight tubes. - It has been found that experimental determinations of the values of h for turbulent flow through straight tubes with cross sections of any shape generally can be correlated by the single equation

$$\frac{hD}{k} = C_1 \left(\frac{c_p \mu g}{k}\right)^l \left(\frac{\rho V D}{\mu}\right)^m \tag{5}$$

where

$\frac{hD}{k}$ Nusselt number, dimensionless

$\frac{c_p \mu g}{k}$ Prandtl number, dimensionless

and, as before,

$\frac{\rho V D}{\mu}$ Reynolds number, dimensionless

For the turbulent flow of air through smooth straight tubes, the best values for the exponents l and m and the constant C_1 in equation (5) for both heating and cooling, as given in reference 1, are

$$C_1 = 0.023$$

$$l = 0.4$$

$$m = 0.8$$

Substitution of these values in equation (5) gives

$$\frac{hD}{k} = 0.023 \left(\frac{c_p \mu g}{k}\right)^{0.4} \left(\frac{\rho VD}{\mu}\right)^{0.8} \tag{6}$$

For air, $\frac{c_p \mu g}{k} = 0.76$ and may be considered constant with respect to temperature and pressure. Equation (6) then reduces to

$$\frac{hD}{k} = 0.02 \left(\frac{\rho VD}{\mu}\right)^{0.8} \tag{7}$$

Because, for the units used in the present report, the numerical value of k is 10 times the numerical value of μ for air at any given temperature, the equation further simplifies to

$$h = 0.2 \left(\frac{\mu}{D}\right)^{0.2} (\rho V)^{0.8} \tag{8}$$

(The constant 0.2 obviously is not dimensionless.)

For the steady flow of air through a tube of constant hydraulic diameter and constant cross-sectional area, the quantities D and ρV in equation (8) are constants throughout the length of the tube. The coefficient of viscosity μ is a function of temperature alone. The variation of $\mu^{0.2}$ throughout the length of a tube of a heat exchanger is very small. For the turbulent flow of air, therefore, the value of h can be considered constant throughout the length of the tube.

Problem:

Calculate the heat-transfer coefficient h for the flow of air through a circular tube 1/4 inch in diameter. The initial air temperature is 0° F and the final temperature is 75° F. The initial density is 0.0022 slug per cubic foot and the initial velocity is 100 feet per second.

Solution:

The value of h is found by using equation (8),

$$h = 0.2 \left(\frac{\mu}{D}\right)^{0.2} (\rho V)^{0.8} \text{ Btu/sec/sq ft/}^{\circ}\text{F} \tag{8}$$

The value of μ at the average air temperature of 37.5° F, as found from figure 67 of appendix A, is

$$\mu = 3.6 \times 10^{-7} \text{ slug/ft-sec}$$

The initial value of ρV can be used because ρV is a constant throughout the length of the tube. Then,

$$h = 0.2 \left(\frac{3.6 \times 48}{10^7}\right)^{0.2} (2.2 \times 10^{-3} \times 10^2)^{0.8}$$

$$= 0.2 \times 0.11 \times 0.3$$

$$= 6.6 \times 10^{-3} \text{ Btu/sec/sq ft/}^{\circ}\text{F}$$

Problem:

Calculate the weight rate of flow W for 250 of the tubes in the preceding problem.

Solution:

$$W = gA\rho Vn$$

$$= 32.2 \times \frac{\pi}{4} \times \left(\frac{1}{48}\right)^2 \times 0.0022 \times 100 \times 250$$

$$= 0.60 \text{ lb/sec}$$

The transition region. - Between the entrance to a tube and the region in which the flow in the tube has the characteristics of fully developed turbulent flow, there is a region of transition in which the velocity distribution across the tube is changing from a uniform distribution at the entrance of the tube to the distribution that characterizes turbulent flow. (See fig. 2.) In the transition

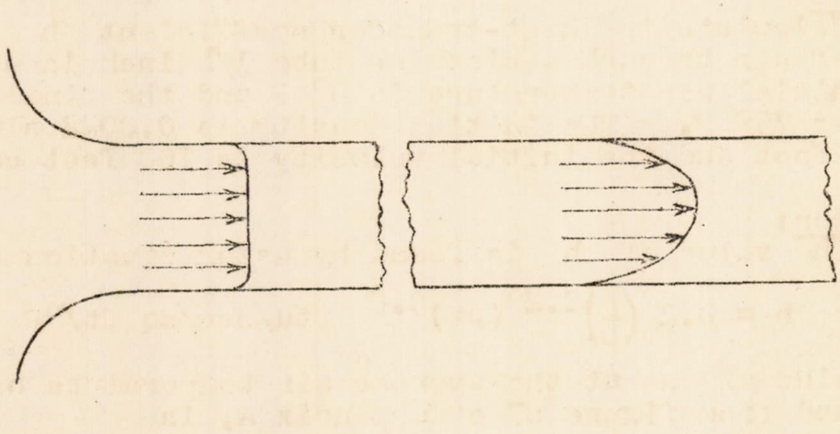

Figure 2. - Velocity distribution near tube entrance and velocity distribution of turbulent flow.

region, the value of h is somewhat greater than the value given by equation (6), which applies to fully established turbulent flow. It is advisable, however, to use the smaller value belonging to fully developed turbulent flow, as given by equation (6), in making calculations for the types of heat exchanger now in common use. Experience has shown that, with the Reynolds numbers and the tube lengths usually employed in heat exchangers, the effect of the transition region can be neglected and equation (6) can be used for calculating h.

Turbulent flow across tube banks. - Experimental determinations of h for the turbulent flow of air across banks of circular tubes can also be correlated by equation (5), rewritten to read

$$\frac{hD}{k} = C_2 \left(\frac{\rho V_{max} D}{\mu} \right)^{0.6} \tag{9}$$

In equation (9), Reynolds number is based on the velocity of the air at the minimum free area V_{max} and on the outside tube diameter D. The exponent m of equation (5) equals 0.6 and the quantity $C_1 \left(\frac{c_p \mu g}{k} \right)^l$ has been replaced by C_2. The value of C_2 depends on: (1) the ratio of the center-to-center spacing of the tubes parallel to the direction of the air flow to the outside diameter of the tubes m_p/D, (2) the ratio of the center-to-center spacing of the tubes normal to the direction of the air flow to the outside diameter of the tubes m_n/D, and (3) the arrangement of the tubes, that is, whether they are in line or staggered.

The values of C_2 as a function of tube spacing have been calculated from the data of reference 2 and are given for in-line tubes in figure 3 and for staggered tubes in figure 4. The data of figures 3 and 4 are applicable only when the tube bank is at least 10 rows deep; they are most nearly accurate for a Reynolds number of 8000 and, for most purposes, are sufficiently accurate over a fairly wide range of Reynolds numbers.

Problem:

The tubes of a heat exchanger in which the cooling air flows across the tubes are so spaced that the center-to-center distance between tubes in the direction normal to the air flow m_n is 0.45 inch and in the direction parallel to the air flow m_p is 0.35 inch. The outside tube diameter D is 0.25 inch, the average fluid temperature is 50° F, and the average fluid pressure is 20 inches of mercury. The

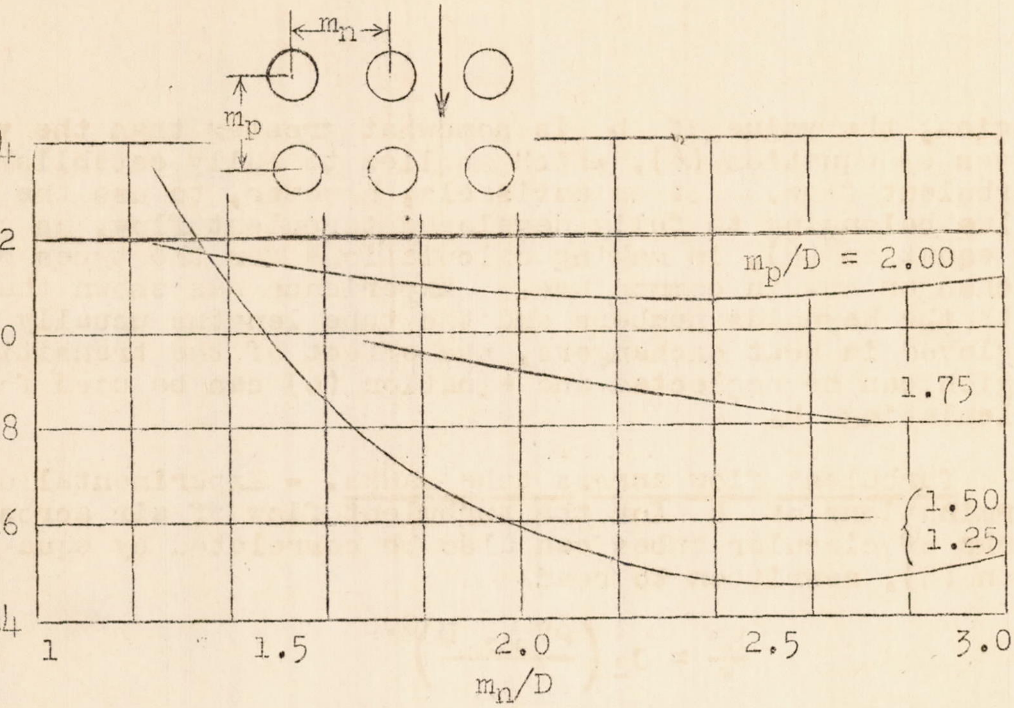

Figure 3. - Constant C_2 as a function of tube spacing for flow of air across in-line tubes. (Data from reference 2.)

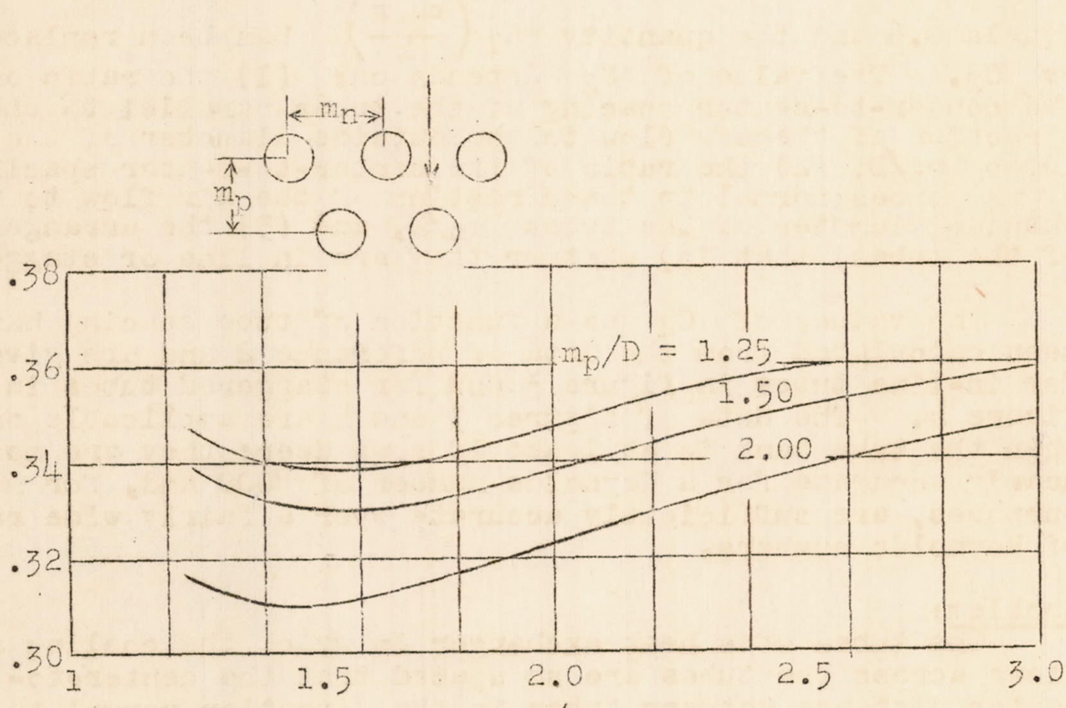

Figure 4. - Constant C_2 as a function of tube spacing for flow of air across staggered tubes. (Data from reference 2.)

velocity of the air at the minimum cross-section V_{max} is measured as 65 feet per second. What is the value of the heat-transfer coefficient h (a) if the tubes are staggered as shown in figure 4 and (b) if the tubes are in line as shown in figure 3?

Solution:

(a) The heat-transfer coefficient h is found by

$$h = \frac{k}{D} C_2 \left(\frac{\rho V_{max} D}{\mu}\right)^{0.6} \quad \text{Btu/sec/sq ft/}^{\circ}\text{F} \qquad (9)$$

From figure 67,

$$k = 3.67 \times 10^{-6} \quad \text{Btu/sec/sq ft/}^{\circ}\text{F/ft}$$

and

$$\mu = 3.67 \times 10^{-7} \quad \text{slug/ft-sec}$$

For the staggered tubes, from figure 4, for $m_n/D = 0.45/0.25 = 1.8$ and $m_p/D = 0.35/0.25 = 1.4$,

$$C_2 = 0.335$$

The density ρ is found as follows: The density of NACA standard air under sea-level conditions - pressure, 29.92 inches of mercury; temperature, 59° F - is 0.002378 slug per cubic foot. (See appendix B.) The density at a pressure of 20 inches of mercury at 50° F is therefore

$$\rho = 0.002378 \times \frac{20}{29.921} \times \frac{518}{509}$$

$$= 0.00162 \quad \text{slug/cu ft}$$

If this value is substituted in equation (9),

$$h = 3.67 \times 10^{-6} \times 48 \times 0.335 \left(\frac{0.00162 \times 65 \times 10^{7}}{48 \times 3.67}\right)^{0.6}$$

$$= 1.1 \times 10^{-2} \quad \text{Btu/sec/sq ft/}^{\circ}\text{F}$$

(b) For the in-line tubes, for $m_n/D = 1.8$ and $m_p/D = 1.4$, figure 3 gives

$$C_2 = 0.27$$

Therefore,

$$h = \frac{0.27}{0.335} \times 1.1 \times 10^{-2}$$

$$= 8.9 \times 10^{-3} \text{ Btu/sec/sq ft/}^{\circ}\text{F}$$

Laminar flow through tubes. - In contrast with the case of turbulent flow through tubes, the case of laminar flow through tubes is in an unsatisfactory status. As previously pointed out, the value of h for turbulent flow can be considered constant throughout the length of the tube. For laminar flow, however, the value of h generally is a function of the length of the tube. Equation (2) accordingly becomes

$$H = \frac{S}{L} \int_0^L h_x \, \Delta T \, dx \qquad (10)$$

The integration in equation (10) is relatively difficult. The problem is greatly simplified by using, instead of equation (10), equation (3)

$$H = hS \, \overline{\Delta T} \qquad (3)$$

in which $\overline{\Delta T}$ is the logarithmic-mean temperature difference and h is the corresponding heat-transfer coefficient. Although h does not represent the true mean value of h_x, its use has definite advantages in addition to the simplicity of equation (3). Convenient equations are available for approximating h. The use of h permits comparisons with results for turbulent flow. As L increases, h approaches a lower limit that is the same as the lower limit of h_x.

Fewer experimental data on heat transfer are available for laminar flow than for turbulent flow. The results that are presented in the following paragraphs are theoretical results taken from reference 3. In order to obtain these results, certain simplifying assumptions were made relative to the conditions under which the heat transfer occurs. If it be assumed that both the tube-wall temperature and the fluid properties - c_p, ρ, and k - are constant throughout the length of the tube and that the velocity distribution across the tube is parabolic, the value of h is given by figure 5 (fig. 2 of reference 3). Figure 5 shows the Nusselt number hD/k as a function of Φ, where

$$\Phi = \frac{c_p \mu g}{k} \frac{\rho V D}{\mu} \frac{D}{L}$$

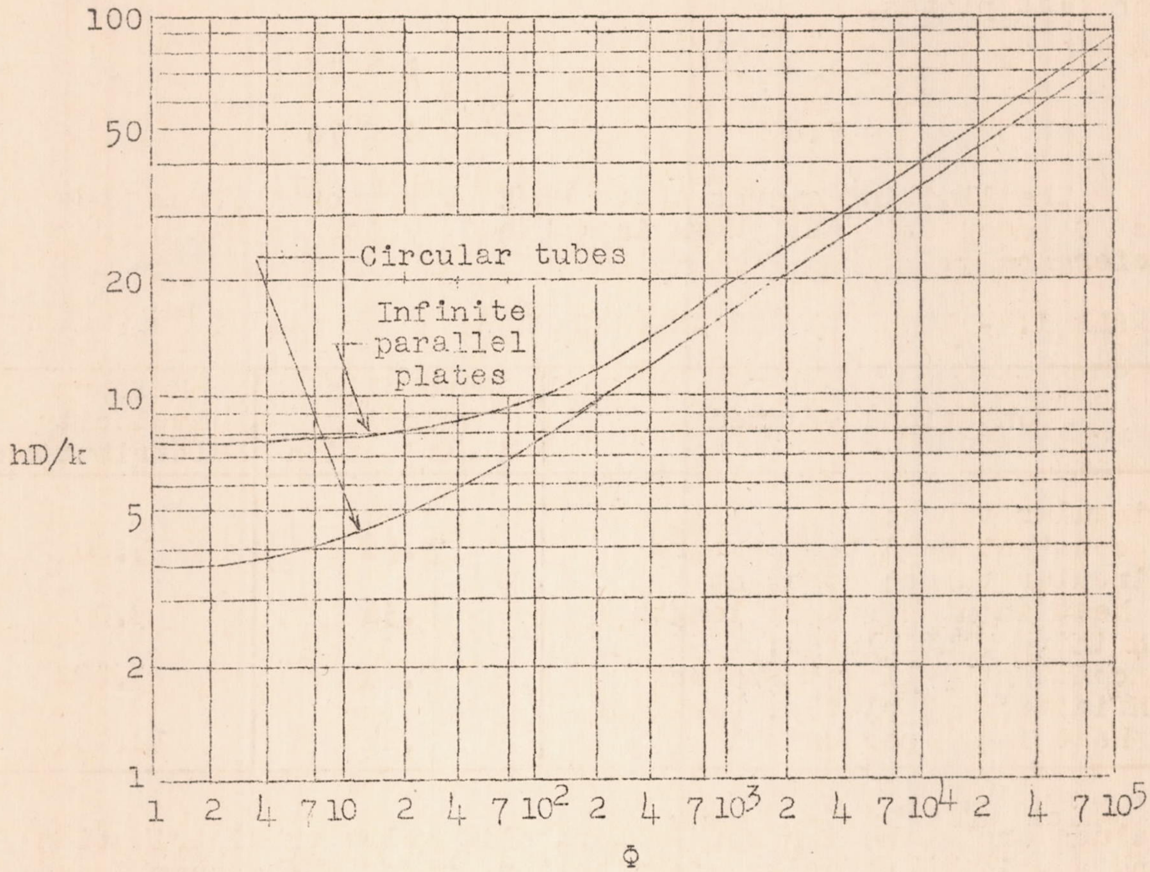

Figure 5. - Theoretical heat-transfer correlation for laminar flow - Nusselt number as a function of Φ.

$$\Phi = \frac{c_p \mu g}{k} \frac{\rho V D}{\mu} \frac{D}{L}. \qquad \text{(From reference 3.)}$$

One curve of figure 5 is for flow through circular tubes; the other is for flow between flat plates of infinite extent perpendicular to the flow direction. It should be noted that h is approximately a constant for small values of Φ.

The curves of figure 5 can be approximated by the following relations:

For circular tubes,

$$\frac{hD}{k} = 1.615\ \Phi^{1/3} \qquad \Phi > 12$$

$$= 3.66 \qquad \Phi < 12$$

For flat plates,

$$\frac{hD}{k} = 1.85\ \Phi^{1/3} \qquad \Phi > 70$$

$$= 7.60 \qquad \Phi < 70$$

The limiting values that hD/k approaches (from above) as Φ decreases are shown in table I, which is taken from reference 3:

TABLE I. - LOWER LIMITS APPROACHED BY hD/k AS Φ DECREASES

Conditions assumed	Parabolic velocity distribution	Uniform velocity distribution
Circular tubes; constant wall temperature	3.66	5.80
Circular tubes; constant heat input per unit length	4.36	8.00
Infinite flat plates; constant wall temperature	7.60	9.88
Infinite flat plates; constant heat input per unit length	8.24	12.00

Values are given for both a parabolic velocity distribution and a uniform velocity distribution, under the assumptions of constant wall temperature and constant heat input per unit length. Generally, none of these conditions is realized in practice. In lieu of more directly applicable results for cases occurring in practice, however, it may be found convenient to estimate, from the data of table I, the value of hD/k for any given set of conditions. For the sake of safety, the lowest of the more nearly applicable values given in the table might be used as the average value of hD/k throughout the tube.

Mean Temperature Difference $\overline{\Delta T}$

When the heat-transfer coefficient h is constant and the temperature of the fluid changes linearly with the amount of heat transferred to or from it, the proper mean temperature difference $\overline{\Delta T}$ for use in equation (3)

$$H = hS\ \overline{\Delta T} \qquad (3)$$

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is the logarithmic-mean temperature difference

$$\overline{\Delta T} = \frac{\Delta T_i - \Delta T_o}{\log_e \frac{\Delta T_i}{\Delta T_o}} \tag{11}$$

All equations and numerical data given in the present report for correlating and calculating values of h are based on this definition of mean temperature difference. Two other definitions of $\overline{\Delta T}$ have been used, however, in the literature: the inlet temperature difference and the arithmetic-mean temperature difference. The convection equations corresponding to these definitions are, respectively,

$$H = h_i S\ \Delta T_i$$

$$H = h_a S\ \overline{\Delta T_a}$$

Although the use of these alternative definitions of temperature difference is permissible, the use of the inlet difference in particular is inadvisable because it does not permit correlation of test results over a wide range of the variables that determine the rate of heat transfer.

In many cases, when the difference between ΔT_i and ΔT_o is not too large a part of ΔT_i, the arithmetic-mean difference defined as

$$\overline{\Delta T_a} = \frac{1}{2}(\Delta T_i + \Delta T_o)$$

can be used, with negligible error, instead of the logarithmic-mean difference. In fact, when ΔT_o is larger than $\frac{1}{2}\Delta T_i$, the error caused by using the arithmetic-mean rather than the logarithmic-mean temperature difference is less than 4 percent.

Consider the case in which part of the heat-transfer surface area is fin area; that is, part of the surface is indirectly heated. The heat that is transferred to the fluid from the directly heated part of the surface traverses only a short path in the metal and therefore experiences a relatively small thermal resistance in the metal; whereas the heat that is transferred from the surface of the fins traverses a relatively long path in the metal and experiences a relatively large thermal resistance. The result is that the surface of the fins operates at a lower temperature than the direct surface. A fin is therefore less effective, per unit surface area, than direct surface for heat transfer.

A fin effectiveness f can be defined as the ratio

$$f = \frac{\Delta T_f}{\Delta T_d}$$

where ΔT_f is the temperature difference between the fin and the fluid and ΔT_d is the temperature difference between the unfinned part of the surface at the base of the fin and the fluid. For this case, equation (3) becomes

$$H = hS_d\,\overline{\Delta T} + hS_f\,\overline{\Delta T}\,f$$

$$= h\,(S_d + S_f f)\,\overline{\Delta T} \qquad (12)$$

in which $\overline{\Delta T}$ is based on the temperature of the direct surface.

By using a relation developed in reference 4 it can be shown that, for engine cylinders with circumferential fins, equation (12) becomes (see fig. 6)

$$H = h\,\frac{S_b}{s + t}\left[\left(2w + \frac{w^2}{r_b}\right) f + s\right]\overline{\Delta T} \qquad (13)$$

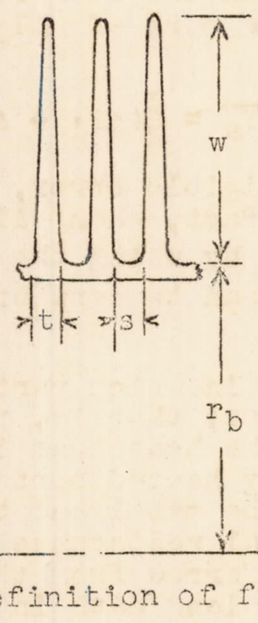

Figure 6. - Definition of fin symbols.

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For straight fins of rectangular cross section, the value of the fin effectiveness (reference 5) is

$$f = \frac{\tanh aw}{aw} \tag{14}$$

where

$$a = \sqrt{\frac{2h}{k_m t}}$$

Equation (14) can also be used with negligible error for the rectangular or the tapered circumferential fins on an air-cooled engine cylinder (reference 5).

A simpler expression for f is the approximate equation (reference 6)

$$f = 1.07 - 0.3\ aw \tag{15}$$

The value of f from equation (15) differs from the value from equation (14) by not more than 1 percent for $0.50 < f < 0.95$.

Applications

Heat transfer in the radiator. - Application of the material that has been presented can now be made in actual calculations of rate of heat transfer. Consider first the case of heat transfer from the wall of a circular tube to air flowing through the tube. Let the tube wall be kept at a constant temperature throughout its length. Although the case of constant tube-wall temperature may at first appear to be one not encountered in aircraft heat exchangers, actually it is the case of the aircraft coolant radiator. The water or the mixture of water and ethylene glycol used in radiators keeps the entire cooling surface at essentially the same temperature, which is very nearly the average temperature of the coolant.

It is desired to be able to calculate the rate of heat transfer from a tube of given dimensions when only the tube-wall temperature and the physical properties and the velocity of the air are known. As was pointed out previously, the rate of heat transfer from any surface is such that both the convection equation

$$H = hS\ \overline{\Delta T} \tag{3}$$

and the heat-balance equation

$$H = W\ c_p\ (T_o - T_i) \tag{4}$$

are satisfied. It is convenient to combine equations (3) and (4) into a single expression for H that does not contain the unknown T_o as follows: If H is eliminated between equations (3) and (4) and substitution is made for $\overline{\Delta T}$ from equation (11),

$$\frac{hS}{W\ c_p} = \log_e \frac{T_w - T_i}{T_w - T_o} \tag{16}$$

(Assuming counterflow or constant wall temp)

Equation (16) is easily transformed into the expression

$$T_o - T_i = (T_w - T_i)\left(1 - e^{-\frac{hS}{W\ c_p}}\right) \tag{17}$$

If equation (17) is substituted in equation (4),

$$H = W\ c_p\ (T_w - T_i)\left(1 - e^{-\frac{hS}{W\ c_p}}\right) \tag{18}$$

If a substitution for h is made from equation (8) and if

$$S = \pi DLn$$

$$W = gA\rho V$$

$$A = \frac{\pi}{4} D^2\ n$$

and

$$\Delta T_i = T_w - T_i$$

are used, the heat-transfer equation for use in radiator calculations is obtained

Assumptions: Counterflow $h \sim R^{.8}$

$$H = W\ c_p\ \Delta T_i \left(1 - e^{-0.1R^{-0.2}\frac{L}{D}}\right) \tag{19}$$

$$= W\ c_p\ \Delta T_i\ \eta$$

It is interesting to show graphically the relation between η, where

$$\eta = \frac{\Delta T_{air}}{\Delta T_i} = \frac{T_o - T_i}{T_w - T_i} = 1 - e^{-0.1R^{-0.2}\frac{L}{D}} \qquad (20)$$

and Reynolds number and L/D. This relation is shown in figure 7. Figure 7 shows the manner in which the final

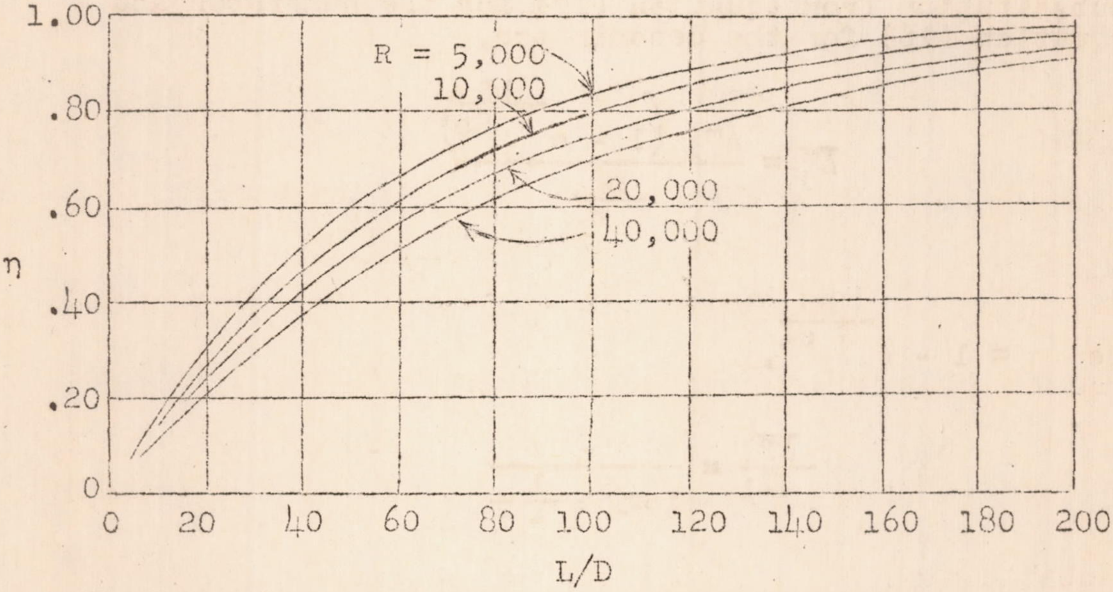

Figure 7. - Variation of η with tube length-diameter ratio and Reynolds number.

temperature of the air approaches the temperature of the wall as the value of L/D is increased.

For small values of L/D, η is comparatively small. For small values of L/D, therefore, the temperature of the air is increased not much beyond its initial value. The mean temperature difference between wall and air is greater therefore for small values than for large values of L/D. The expression for the mean temperature difference is easily obtained, as follows: The mean temperature difference $\overline{\Delta T}$ is defined by equation (11) as

$$\overline{\Delta T} = \frac{\Delta T_i - \Delta T_o}{\log_e \frac{\Delta T_i}{\Delta T_o}} \qquad (11)$$

counterflow

For a constant wall temperature, the equation becomes

$$\overline{\Delta T} = \frac{T_o - T_i}{\log_e \frac{\Delta T_i}{\Delta T_o}}$$

By a substitution from equation (17) for the numerator and from equation (16) for the denominator,

$$\overline{\Delta T} = \frac{\Delta T_i \left(1 - e^{-\frac{hS}{W\ c_p}}\right)}{\frac{hS}{W\ c_p}}$$

Because $\eta = 1 - e^{-\frac{hS}{W\ c_p}}$,

$$\frac{\overline{\Delta T}}{\Delta T_i} = \frac{\eta}{\log_e \frac{1}{1 - \eta}}$$

Inasmuch as η can be expressed as a function of L/D and Reynolds number alone,

$$\eta = 1 - e^{-0.1R^{-0.2} \frac{L}{D}}$$

the dimensionless mean temperature difference $\overline{\Delta T}/\Delta T_i$ can also be expressed as

$$\frac{\overline{\Delta T}}{\Delta T_i} = \frac{1 - e^{-0.1R^{-0.2} \frac{L}{D}}}{0.1R^{-0.2} \frac{L}{D}} \tag{21}$$

Equation (21) is plotted in figure 8, which shows small values of L/D to be advantageous on account of the larger values of mean temperature difference. Heat exchangers, however, are designed to have not the lowest values of L/D. The use of short tubes and tubes of large diameter results in heat exchangers that have large frontal areas.

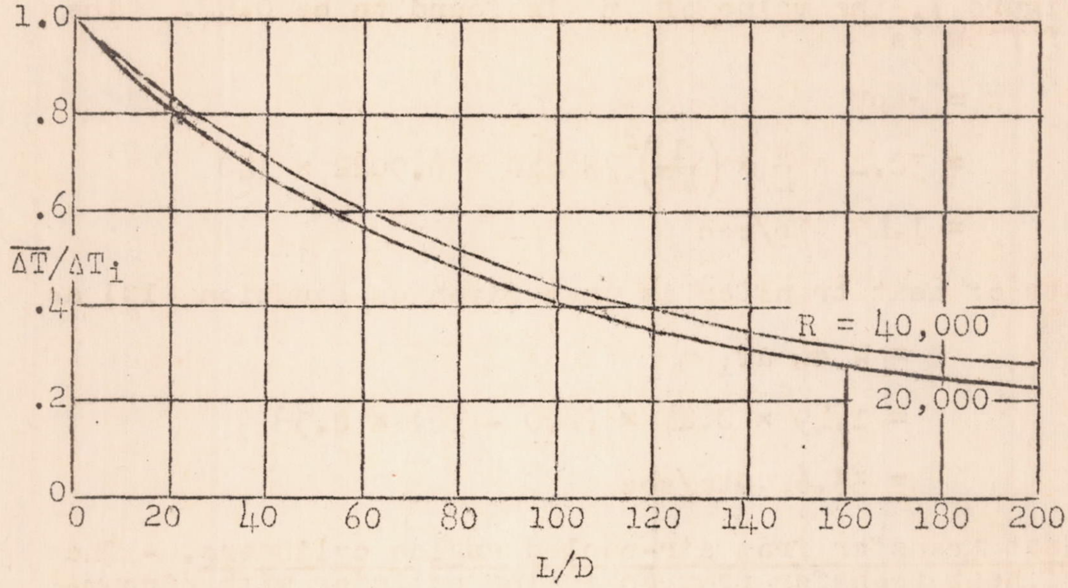

Figure 8. - Variation of $\overline{\Delta T}/\Delta T_i$ with tube length-diameter ratio and Reynolds number.

Problem:

What is the rate of heat transfer in a radiator composed of 440 tubes that are 15 inches long and 1/4 inch in inside diameter? The tube-wall temperature is kept essentially constant throughout the radiator at 200° F. The cooling air flows through the tubes and has a coefficient of viscosity μ of 3.6×10^{-7} slug per foot-second, a density ρ of 0.0022 slug per cubic foot, a velocity V of 140 feet per second, and an initial temperature of 38° F.

Solution:

The Reynolds number of the flow is

$$\frac{\rho VD}{\mu} = \frac{0.0022 \times 140}{3.6 \times 10^{-7} \times 48}$$

$$= 17,800$$

The length-diameter ratio of the tubes is

$$\frac{L}{D} = 15 \times 4$$

$$= 60$$

From figure 7, the value of η is found to be 0.58. The rate of flow is

$$
\begin{aligned}
W &= gA\rho V \\
&= 32.2 \times \frac{\pi}{4} \times \left(\frac{1}{48}\right)^2 \times 440 \times 0.0022 \times 140 \\
&= 1.49 \text{ lb/sec}
\end{aligned}
$$

The rate of heat transfer is then given by equation (19) as

$$
\begin{aligned}
H &= W\, c_p\, \Delta T_i\, \eta \\
&= 1.49 \times 0.24 \times (200 - 38) \times 0.58 \\
&= 33.6 \text{ Btu/sec}
\end{aligned}
$$

Heat transfer from air-cooled engine cylinders. - The rate of heat transfer from an engine cylinder with circumferential fins to the cooling air can be calculated by equations (13) and (4). The quantity $\overline{\Delta T}$ is properly the logarithmic-mean temperature difference, although the arithmetic-mean difference can be used with small error. (See section entitled "Mean Temperature Difference $\overline{\Delta T}$.") In the calculation of the effect of changes in the fin dimensions on the rate of heat transfer or on the weight rate of flow required to cool a cylinder, equation (8) is sufficiently accurate for determining the coefficient h.

HEAT TRANSFER - BETWEEN TWO FLUIDS

All the foregoing discussion of part I applies to the transfer of heat between a fluid and a surface. In heat exchangers, the heat transfer takes place between two fluids separated by a wall. The temperature gradient between the two fluids at a representative point is shown in figure 9. The discussion of the simpler case of heat transfer between a fluid and a solid is pertinent because the heat exchanger is simply an extension of this case in which there are two combinations of fluid and surface.

Basic Equations

As in the case of heat transfer between wall and fluid, the rate of heat transfer in a heat exchanger is determined by an energy-balance equation and a convection equation.

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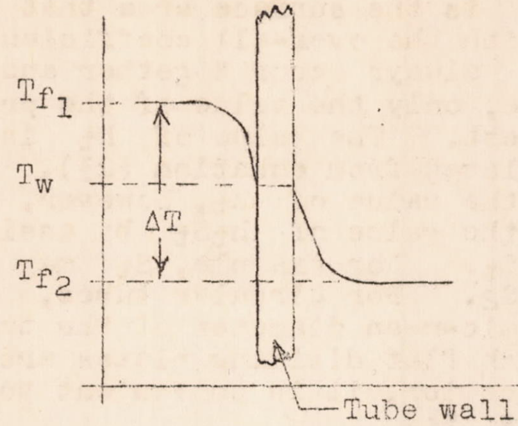


Figure 9. - Temperature gradient between the two fluids in a heat exchanger.

The energy-balance equation applies to each of the two fluids in the heat exchanger. For the cold fluid,

$$H = W_c \, c_p \, (T_o - T_i)_c$$

For the hot fluid,

$$H = W_h \, c_p \, (T_i - T_o)_h$$

The convection equation, which is similar to equation (3), is

$$H = h_t S_t \, \overline{\Delta T} \tag{22}$$

The quantity h_t is the over-all coefficient of heat transfer between the two fluids and is a function of too many quantities for a simple correlation equation, such as equation (5), to exist. The value of h_t, however, can be calculated by use of the following equation:

$$\frac{1}{h_t S_t} = \frac{1}{h_1 S_1} + \frac{1}{h_2 S_2} \tag{23}$$

In equation (23), h_1 is the coefficient of heat transfer between one fluid and the surface S_1 in contact with it, and h_2 is the coefficient for the other fluid and the surface S_2 in contact with it. Each of these coefficients can be calculated by the equations given in preceding sections. The values of S_1 and S_2 generally are different, because one may be based on the inside diameter of a tube and the other on the outside diameter or one may be based on a finned surface and the other on an unfinned surface.

The quantity S_t is the surface area that is to be used in conjunction with the over-all coefficient h_t. Actually, h_t and S_t always occur together and always as a product and, therefore, only the value of the product h_tS_t generally is of interest. The value of h_t is not measured directly but is calculated from equation (23). Should it be necessary to know the value of h_t, however, its value can be obtained from the value of h_tS_t by assigning any convenient value to S_t. For example, S_t may be taken equal to S_1 or to S_2. For circular tubes, S_t may be based on the logarithmic-mean diameter of the tube. In a heat exchanger in which flat dividing plates are used, such as the Harrison intercooler, it is convenient to take the dividing-plate area as S_t.

Equation (23) is a resistance equation, in which $1/h_1S_1$ is the thermal resistance between one fluid and its metal surface, $1/h_2S_2$ is the thermal resistance between the other fluid and its metal surface, and $1/h_tS_t$ is the total or over-all thermal resistance between the two fluids. Actually, the right-hand side of equation (23) should contain a third term t/k_mS_t, which represents the thermal resistance of the metal wall between surfaces 1 and 2. For metal walls, however, the magnitude of this term is so small in comparison with the other two terms that the term can be omitted from the equation.

Mean Temperature Difference

The present section is a discussion of the calculation of the mean temperature difference between the two fluids in a heat exchanger, that is, of the average value throughout the entire heat exchanger of the ΔT shown in figure 9. In this discussion it is convenient to use nondimensional quantities. The three quantities to be considered here - drop in temperature of the hot fluid, rise in temperature of the cold fluid, and mean temperature difference between the two fluids - are made dimensionless by dividing by ΔT_i, the difference in the inlet temperatures of the hot fluid and the cold fluid. Thus,

$$\xi = \frac{\Delta T_h}{\Delta T_i} = \frac{\text{Drop in temperature of hot fluid}}{\text{Difference in inlet temperatures}}$$

$$\eta = \frac{\Delta T_c}{\Delta T_i} = \frac{\text{Rise in temperature of cold fluid}}{\text{Difference in inlet temperatures}}$$

$$\zeta = \frac{\overline{\Delta T}}{\Delta T_i} = \frac{\text{Mean temperature difference}}{\text{Difference in inlet temperatures}}$$

Equation (22) may then be written

$$H = h_t S_t \zeta \, \Delta T_i \tag{24}$$

Equation (4) may be written, for the hot fluid, as

$$H = W_h \, c_p \xi \, \Delta T_i \tag{25}$$

and, for the cold fluid, as

$$H = W_c \, c_p \eta \, \Delta T_i \tag{26}$$

The two fluids in a heat exchanger may flow parallel in the same direction (parallel flow), parallel in opposite directions (counterflow), or at right angles to each other (crossflow). In parallel-flow and counterflow exchangers, the temperature difference between the two fluids varies along the length of the exchanger, as shown in figure 10. For counterflow, the mean temperature difference between the two fluids in nondimensional form is

$$\zeta = \frac{\xi - \eta}{\log_e \dfrac{1 - \eta}{1 - \xi}} \tag{27}$$

For parallel flow, the mean temperature difference in nondimensional form is

$$\zeta = \frac{\xi + \eta}{\log_e \dfrac{1}{1 - \xi - \eta}} \tag{28}$$

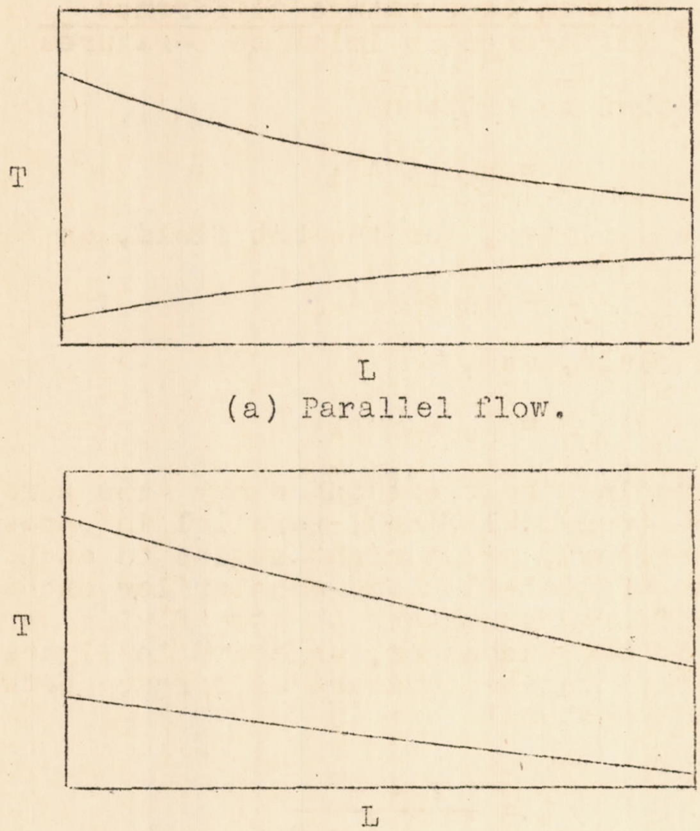
For the parallel-flow heat exchanger, the sum of ξ and η cannot exceed unity. For given values of ξ and η, ζ is less for parallel flow than for either crossflow or counterflow.

Example:

As illustrations of the calculation of ζ, consider heat exchangers in which the hot air enters at 300° F and leaves at 100° F and the cooling air enters at 10° F and leaves at 90° F. The inlet temperature difference is

$$\Delta T_i = 300 - 10$$

$$= 290^\circ \text{ F}$$



(b) Counterflow.

Figure 10. - Temperature gradients in parallel-flow and counterflow heat exchangers.

The drop in the temperature of the hot fluid is

$$\Delta T_h = 300 - 100$$
$$= 200^\circ \text{ F}$$

Then,

$$\xi = \frac{\Delta T_h}{\Delta T_i}$$
$$= \frac{200}{290}$$
$$= 0.69$$

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The rise in the temperature of the cold fluid is

$$\Delta T_c = 90 - 10$$
$$= 80^\circ \text{ F}$$

Then,

$$\eta = \frac{\Delta T_c}{\Delta T_i}$$
$$= \frac{80}{290}$$
$$= 0.28$$

For counterflow, by equation (27),

$$\zeta = \frac{\xi - \eta}{\log_e \frac{1 - \eta}{1 - \xi}} \tag{27}$$

$$= \frac{0.69 - 0.28}{\log_e \frac{0.72}{0.31}}$$

$$= \frac{0.41}{0.84}$$

$$= 0.49$$

For parallel flow, by equation (28),

$$\zeta = \frac{\xi + \eta}{\log_e \frac{1}{1 - \xi - \eta}} \tag{28}$$

$$= \frac{0.69 + 0.28}{\log_e \frac{1}{1 - 0.69 - 0.28}}$$

$$= \frac{0.97}{\log_e \frac{1}{0.03}}$$

$$= 0.28$$

In a crossflow exchanger, the temperature difference between the fluids varies not only along the length of the exchanger but also across the width. Figure 11 (taken from figures 3 to 5 of reference 7) shows the distribution throughout a crossflow heat exchanger of the temperature of the hot fluid, the temperature of the cold fluid, and the

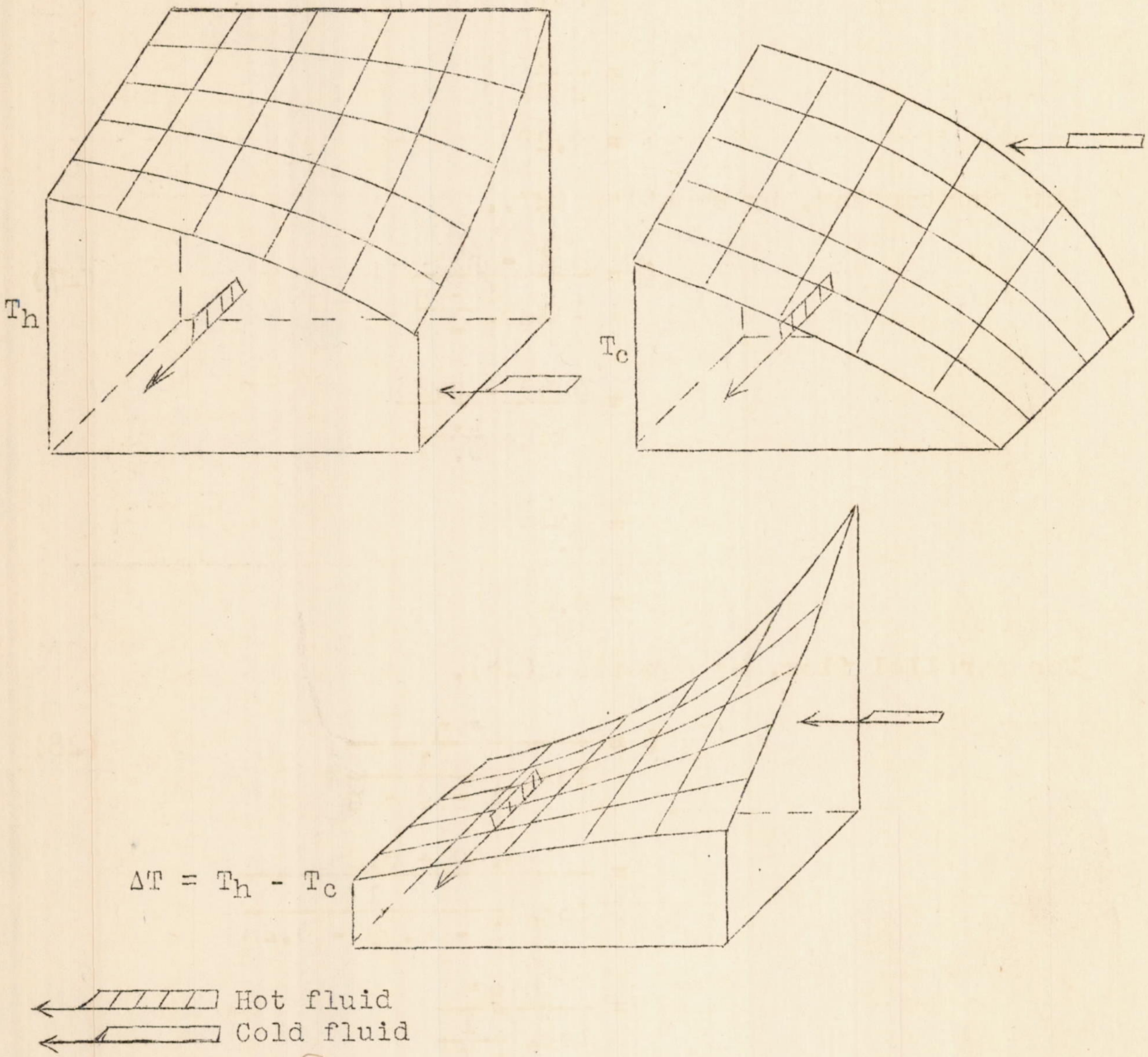


Figure 11. - Distribution in a crossflow heat exchanger of the temperature of the hot fluid, the temperature of the cold fluid, and the temperature difference between the two fluids. (From figs. 3 to 5 of reference 7.)

temperature difference between the two fluids for a representative set of conditions. The quantity $\overline{\Delta T} = \zeta\, \Delta T_i$ is the mean value, throughout the exchanger, of the temperature difference between the two fluids. For the crossflow exchanger, there is no simple exact expression relating ζ, ξ, and η. The values of ζ for various values of ξ and η have been calculated by Nusselt (reference 8) and are presented in table II. The data of table II are plotted in figure 12 with $1/\zeta$ as a function of $W_c c_{p_c}/W_e c_{p_e}$ and ξ.

TABLE II. - VALUES OF ζ FOR CROSSFLOW (FROM REFERENCE 8)

ξ \ η	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
0	1.000	0.947	0.893	0.838	0.781	0.721	0.657	0.586	0.502	0.388	0
.1	.947	.893	.840	.786	.729	.670	.605	.533	.448	.338	0
.2	.893	.840	.785	.734	.677	.617	.552	.480	.398	.292	0
.3	.838	.786	.734	.682	.625	.565	.502	.430	.348	.247	0
.4	.781	.729	.677	.625	.569	.513	.449	.378	.300	.206	0
.5	.721	.670	.617	.565	.513	.456	.394	.326	.251	.167	0
.6	.657	.605	.552	.502	.449	.394	.334	.271	.201	.128	0
.7	.586	.533	.480	.430	.378	.326	.271	.213	.151	.089	0
.8	.502	.448	.398	.348	.300	.251	.201	.151	.100	.052	0
.9	.388	.338	.292	.247	.206	.167	.128	.089	.052	.022	0
1.0	0	0	0	0	0	0	0	0	0	0	0

Problem:

What is the value of ζ for a crossflow heat exchanger for the values of ξ and η used in the preceding example?

Solution:

For $\xi = 0.69$ and $\eta = 0.28$, table II gives $\zeta = 0.447$.

Figure 12 can also be used to find ζ. Because

$$\frac{W_c\, c_{p_c}}{W_e\, c_{p_e}} = \frac{\xi}{\eta}$$

$$= \frac{0.69}{0.28}$$

$$= 2.46$$

the abscissa is 2.46. Therefore,

$$\frac{1}{\zeta} = 2.24$$

and

$$\zeta = 0.447$$

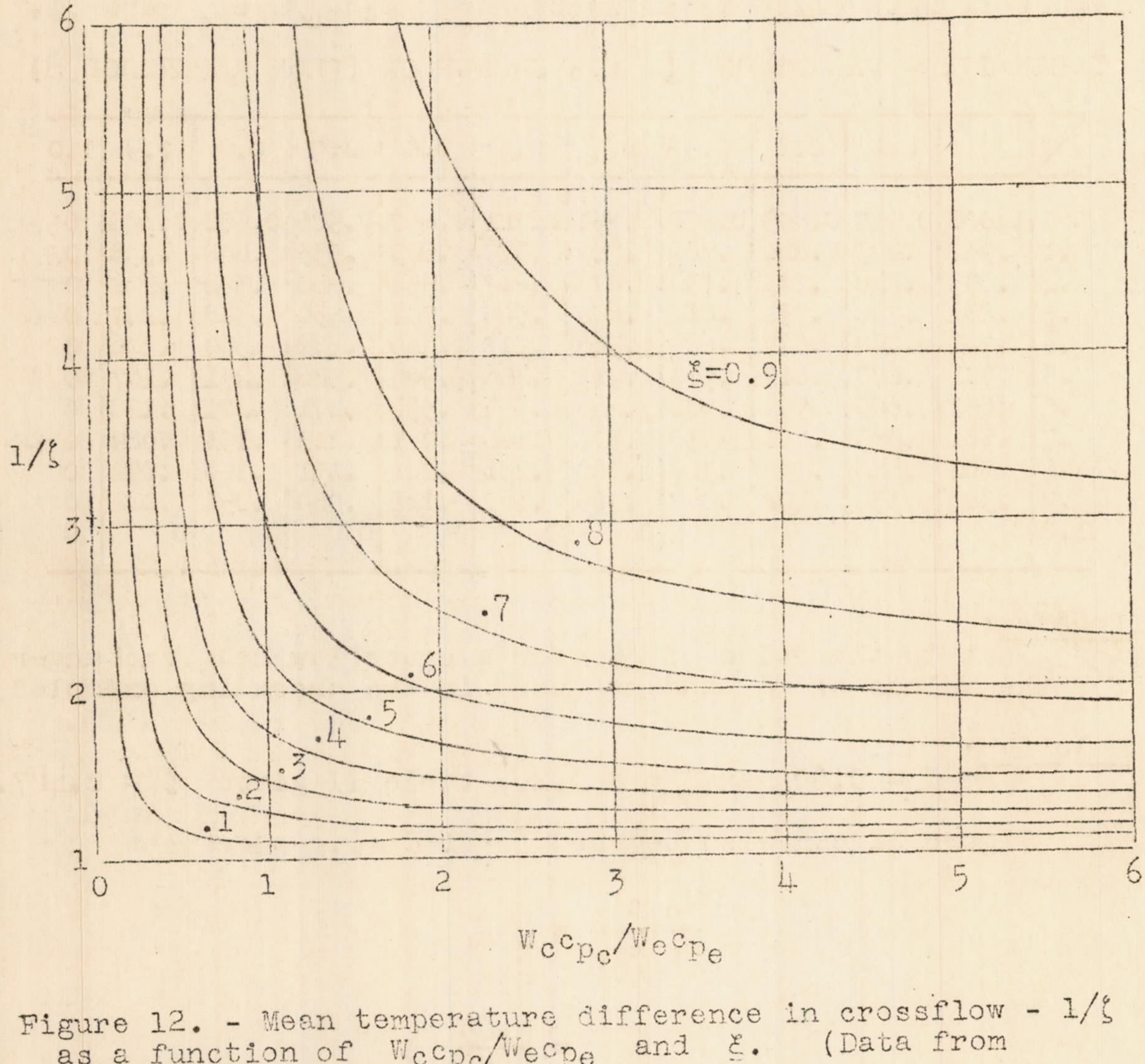


Figure 12. - Mean temperature difference in crossflow - $1/\zeta$ as a function of $W_c c_{p_c}/W_e c_{p_e}$ and ξ. (Data from reference 8.)

The values of the ratio of ζ for crossflow to ζ for counterflow for various values of ξ and η are shown in table III, which is taken from reference 8. It is clear that, for given values of ξ and η, ζ is greater for

TABLE III. - RATIO OF ζ FOR CROSSFLOW TO ζ

FOR COUNTERFLOW (FROM REFERENCE 8)

ξ \ η	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	1	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
.1	1	.996	.994	.992	.988	.984	.978	.973	.961	.937
.2	1	.993	.988	.983	.975	.967	.955	.942	.919	.873
.3	1	.990	.983	.974	.962	.952	.935	.908	.872	.810
.4	1	.987	.975	.962	.948	.935	.909	.873	.824	.738
.5	1	.984	.967	.950	.935	.910	.875	.832	.765	.665
.6	1	.980	.955	.935	.909	.877	.835	.780	.698	.581
.7	1	.975	.942	.911	.875	.832	.780	.710	.614	.485
.8	1	.961	.919	.872	.824	.758	.698	.614	.500	.360
.9	1	.928	.867	.801	.738	.672	.581	.490	.360	.220

counterflow than for crossflow. In other words, the surface area required with counterflow is less than that required with crossflow. It is customary, however, to make aircraft heat exchangers of the crossflow type because it is much simpler to connect manifolding or ducts to the crossflow type than to the counterflow type. (See also section entitled "Applications to Design.")

The assumptions upon which equations (27) and (28) and the data of tables II and III are based should be mentioned. These assumptions are (1) that the temperature of a fluid is a linear function of its heat load and (2) that h_t is constant throughout the heat exchanger. The first assumption is almost always justified. (See equations (4) and (40).) For turbulent flow, the second assumption is justified. For laminar flow, however, the second assumption means that the equations for ζ given in this section are strictly applicable only when Φ (fig. 5) has a value low enough that hD/k is independent of Φ and therefore independent of L.

PRESSURE LOSS

The flow of the cooling air through heat exchangers is accompanied by losses in total pressure. These total-pressure losses result principally from static-pressure losses that are caused by friction and by the increased momentum of the heated air. The loss in total pressure that is caused by the abrupt expansion at the exit from a heat exchanger is discussed in the section on ducts.

Friction Pressure Loss

For both turbulent and laminar flow in straight passages, the drop in static pressure caused by friction is given by the equation

$$\Delta p_f = 4f \frac{L}{D} \frac{\rho V^2}{2} \qquad (29)$$

in which D is the hydraulic diameter of the passage.

The quantity f in equation (29) is known as the friction factor. The value of f is a function of the kind of flow, the roughness of the surface, and the fluid properties and velocity.

For turbulent flow through smooth tubes, f is shown as a function of Reynolds number in figure 13, which is taken

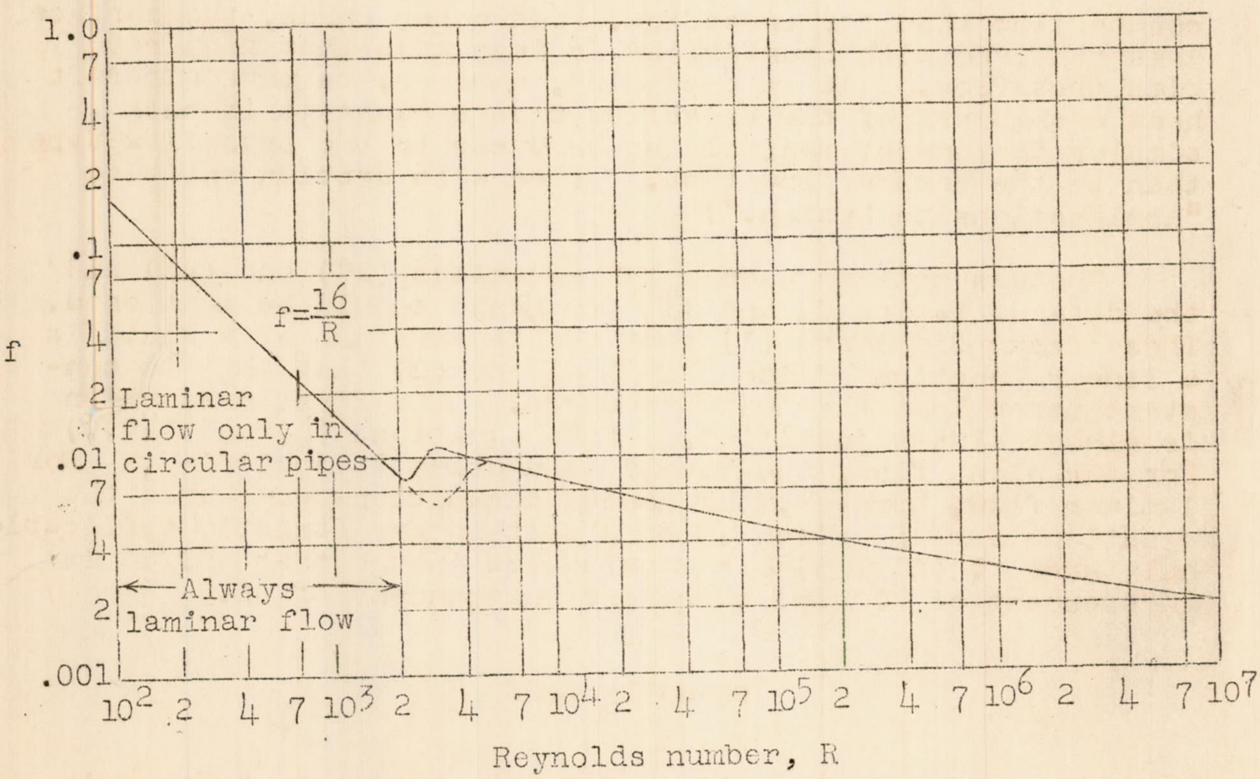


Figure 13. - Friction factor f as a function of Reynolds number R for laminar flow in circular tubes and turbulent flow in smooth tubes with cross sections of any shape. (From fig. 51 of reference 1.)

from figure 51 of reference 1. The part of the curve that applies to turbulent flow is based on many experiments with tubes with cross sections of various shapes. For Reynolds numbers between 5000 and 200,000, the curve of figure 13 is given by the following equation, also taken from reference 1:

$$f = \frac{0.046}{R^{0.2}} \tag{30}$$

For turbulent flow through smooth straight passages over a wide range of Reynolds numbers, the static-pressure drop due to friction can therefore be calculated by the equation

$$\Delta p_f = 0.092\ \rho V^2 \frac{L}{D} R^{-0.2} \tag{31}$$

Problem:

Calculate the friction pressure drop in a tube, 1/4 inch in diameter and 2 feet long, through which air is flowing at 100 feet per second. The average fluid properties in the tube are those of standard sea-level conditions.

Solution:

From appendix B and figure 67, the fluid properties are

$$\rho = 0.002378 \text{ slug/cu ft}$$

$$\mu = 3.73 \times 10^{-7} \text{ slug/ft/sec}$$

The Reynolds number is

$$R = \frac{\rho V D}{\mu}$$

$$= \frac{0.002378 \times 100}{48 \times 3.73 \times 10^{-7}}$$

$$= 13,300$$

By equation (31),

$$\Delta p_f = 0.092\ \rho V^2 \frac{L}{D} \frac{1}{R^{0.2}}$$

$$= \frac{0.092 \times 0.002378 \times 10^4 \times 2 \times 48}{(13300)^{0.2}}$$

$$= 31.4 \text{ lb/sq ft}$$

For laminar flow through smooth straight passages, the value of the friction factor f in equation (29) is

$$f = \frac{C_3}{R} \tag{32}$$

For laminar flow, equation (29) therefore becomes

$$\Delta p_f = 2\ C_3 \mu V \frac{L}{D^2} \tag{33}$$

The calculated values of C_3 for passages with cross sections of various shapes are shown in figure 14, which is taken from reference 9. These theoretical values of C_3 are well supported by experimental data (reference 1).

For the flow of air across banks of circular tubes, the static-pressure drop is given by

$$\Delta p_f = 4f\left(\frac{L}{m_p}\right)\frac{\rho V^2}{2} \tag{34}$$

where L is the length of the tube bank in the direction of flow and L/m_p is the number of tubes on which a particle of fluid can impinge as it flows through the bank, that is, the number of contractions and expansions.

Experimental determinations of the values of f can be correlated by the equation

$$f = \frac{C_4}{R^{0.13}} \tag{35}$$

The value of C_4 depends on the spacing and the arrangement of the tubes, that is, on the ratios m_p/D and m_n/D and on whether the tubes are staggered or in line. The values of C_4 as a function of spacing have been taken from the data of reference 2 and are given in figure 15 for the in-line tubes and in figure 16 for the staggered tubes. The data of figures 15 and 16 are applicable only when the tube bank is at least 10 rows deep; they are most nearly accurate for a Reynolds number of 8000 and, for most purposes, are sufficiently accurate over a fairly wide range of Reynolds numbers.

Problem:

Find the pressure drop across a bank of tubes 0.5 inch in diameter. The tubes are in line with $m_n = 1.0$ inch and $m_p = 0.75$ inch; the bank is 2 feet deep. Air flowing across the bank has the following average properties:

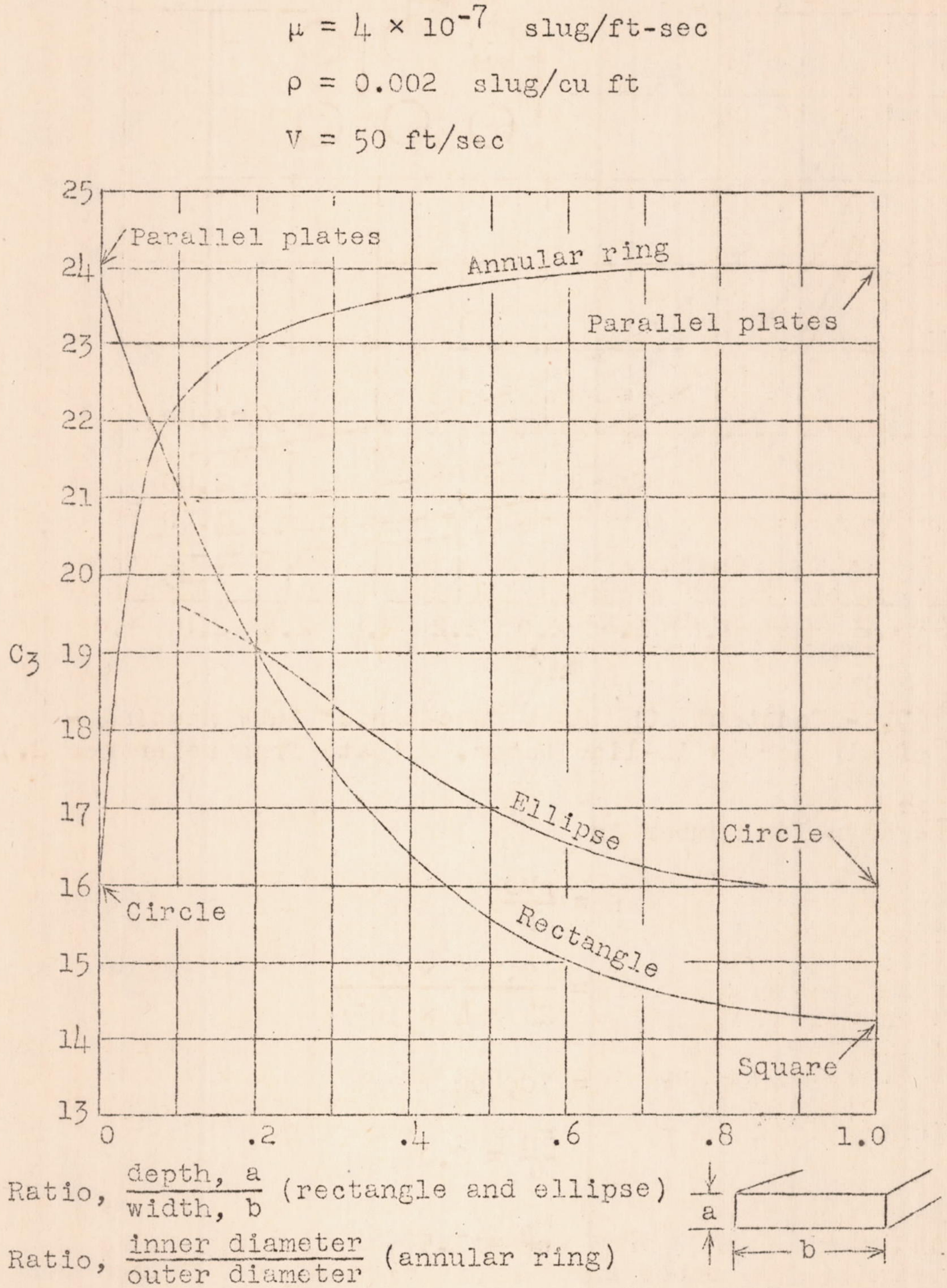


Figure 14. - Proportionality constant C_3 for laminar flow in ducts of various cross-sectional shapes. (Data from reference 9.)

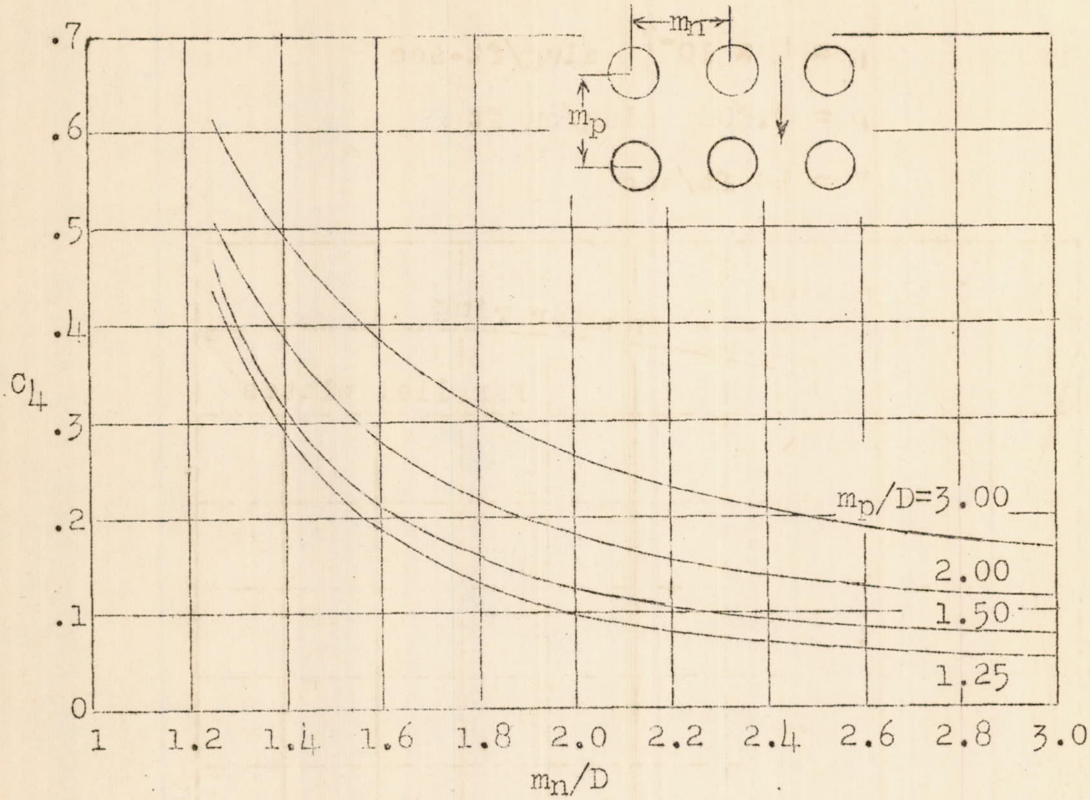


Figure 15. - Constant C_4 as a function of tube spacing - flow of air across in-line tubes. (Data from reference 2.)

Solution:

The Reynolds number is

$$R = \frac{\rho VD}{\mu}$$

$$= \frac{0.002 \times 50}{24 \times 4 \times 10^{-7}}$$

$$= 10,400$$

$$\frac{m_n}{D} = 2.0$$

$$\frac{m_p}{D} = 1.5$$

From figure 15,

$$C_4 = 0.13$$

The friction factor then is

$$f = \frac{C_4}{R^{0.13}}$$

$$= \frac{0.13}{3.33}$$

$$= 0.039$$

By equation (34),

$$\Delta p_f = 2f \frac{L}{m_p} \rho V^2$$

$$= 2 \times 0.039 \times \frac{2}{0.75} \times 12 \times 0.002 \times 2500$$

$$= 12.5 \text{ lb/sq ft}$$

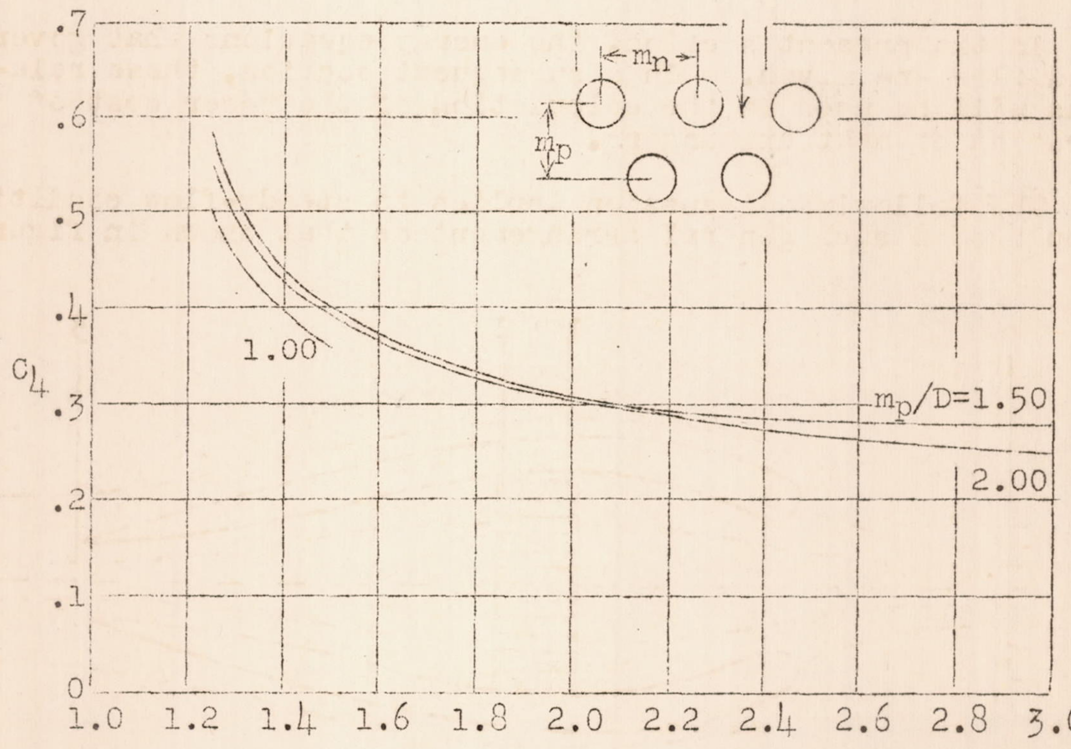


Figure 16. - Constant C_4 as a function of tube spacing - flow of air across staggered tubes. (Data from reference 2.)

Momentum-Increase Pressure Loss

When heat is added to the cooling air that is flowing through a tube of constant cross-sectional area, the density of the air is decreased and its velocity is increased. Because of the increase in velocity, the momentum of the fluid, which is proportional to ρV^2, is increased. The increase in momentum is accompanied by a drop in static pressure, the magnitude of which is given by

$$
\begin{aligned}
\Delta p_m &= \rho_1 V_1 (V_2 - V_1) \\
&= \rho_2 V_2 (V_2 - V_1) \\
&= 2 (q_2 - q_1) \qquad (36)
\end{aligned}
$$

(For the derivation of equation (36), see the derivation of equation (46) in the following section entitled "Energy Balances.")

ENERGY BALANCES

In the present section, the energy equations that govern fluid flow are given. In a subsequent section, these relations will be used in the calculation of the power cost of operation of heat exchangers.

The following discussion applies to steady-flow conditions through some such general arrangement as that shown in figure 17,

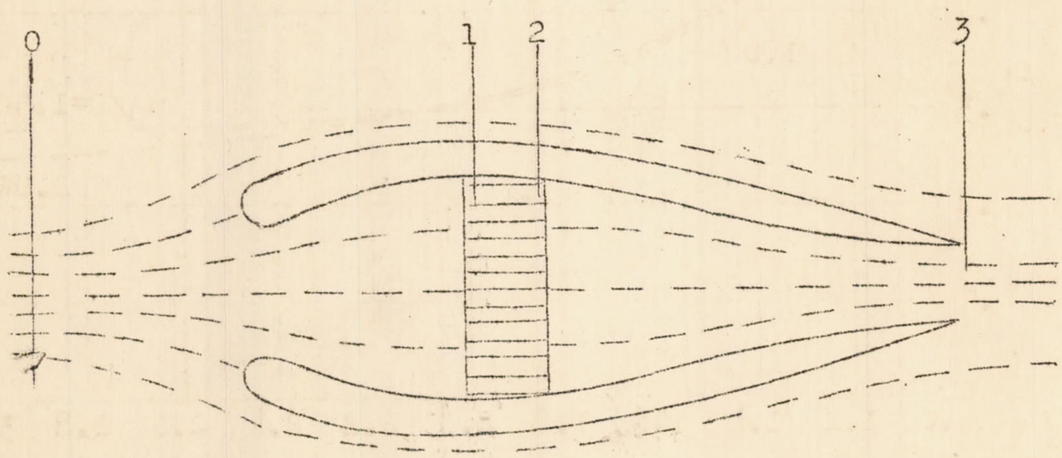


Figure 17. - Schematic representation of heat exchanger in duct.

which represents a heat exchanger installed in a duct. At station 0, free-stream conditions exist. Station 1 is taken just within the heat-exchanger entrance and station 2, just within the heat-exchanger exit. Station 3 is at the exit of the duct.

Total Energy Balance

Consideration can be restricted to the total energy of 1 pound weight of fluid. A useful energy-balance equation can be developed for unit weight of fluid in terms of its properties at stations 1 and 2.

The total energy of unit weight of fluid at any station is its internal or intrinsic energy plus its kinetic energy. The total energy of unit weight at station 2 equals its total energy at station 1 plus the external work done on it plus the heat energy given to it; therefore,

$$JU_2 + \frac{V_2^2}{2g} = JU_1 + \frac{V_1^2}{2g} + J\frac{H}{W} + W_e \tag{37}$$

(Equation (37) is exact for a uniform velocity distribution, that is, for V constant across the tube cross section. For the velocity distribution of turbulent flow in circular tubes, the kinetic energy is about 5 percent to 10 percent greater than $V^2/2g$, where V is the average velocity at the cross section. For the velocity distribution of laminar flow in circular tubes, the kinetic energy is twice $V^2/2g$.) If there is no pump, fan, or turbine between stations 1 and 2, W_e is composed of two parts: The work done on unit weight as it is pushed past station 1 by the fluid behind it and the work done by unit weight in passing station 2 on the fluid ahead of it; that is,

$$W_e = \frac{p_1}{g\rho_1} - \frac{p_2}{g\rho_2} \tag{38}$$

Therefore,

$$JU_2 + \frac{V_2^2}{2g} + \frac{p_2}{g\rho_2} = JU_1 + \frac{V_1^2}{2g} + \frac{p_1}{g\rho_1} + J\frac{H}{W} \tag{39}$$

The difference in internal energy at the two stations is defined by the equation

$$J\ (U_1 - U_2) + \frac{p_1}{g\rho_1} - \frac{p_2}{g\rho_2} = J\ c_p\ (T_1 - T_2)$$

Equation (39) can therefore be written as

$$J\ c_p\ T_2 + \frac{V_2^2}{2g} = J\ c_p\ T_1 + \frac{V_1^2}{2g} + J\ \frac{H}{W} \tag{40}$$

The general gas law

$$pv = \frac{p}{g\rho} = RT$$

and the relation

$$R = J\ (c_p - c_v)$$

can be used in equation (40) to obtain the equation

$$\frac{\gamma}{\gamma - 1}\,\frac{p_2}{\rho_2} + \frac{V_2^2}{2} = \frac{\gamma}{\gamma - 1}\,\frac{p_1}{\rho_1} + \frac{V_1^2}{2} + g\ J\ \frac{H}{W} \tag{41}$$

Equation (41) is the equation for total energy balance and is one form of expression of Bernoulli's well-known theorem of the conservation of energy. Equation (41) is applicable in all cases of fluid flow except those in which the velocity distribution is not constant over a cross section, there is a pump or a turbine in the system, and the general gas law does not hold. It is applicable, for example, when gradual or abrupt changes in cross-sectional area occur between stations 1 and 2.

For air, equation (41) simplifies to

$$7\ \frac{p_2}{\rho_2} + V_2^2 = 7\ \frac{p_1}{\rho_1} + V_1^2 + (5 \times 10^4)\ \frac{H}{W} \tag{42}$$

Mechanical Energy Balance

A useful energy-balance equation can also be written in terms of the mechanical energies involved. The mechanical energy lost per unit weight of fluid by friction - that is,

the mechanical energy which is converted into heat - is called F. The total mechanical work W_m done by the fluid on itself and its surroundings is then given by

$$dW_m = -dF + \frac{p}{g}\, d\left(\frac{1}{\rho}\right)$$

Also, the familiar equation of thermodynamics relating internal energy, heat received, and mechanical work done is

$$dW_m = J\, d\left(\frac{H}{W}\right) - J\, dU$$

Therefore,

$$J\, dU + \frac{p}{g}\, d\left(\frac{1}{\rho}\right) - J\, d\left(\frac{H}{W}\right) = dF$$

Equation (39) can be differentiated to give

$$J\, dU + \frac{p}{g}\, d\left(\frac{1}{\rho}\right) - J\, d\left(\frac{H}{W}\right) = \frac{-dp}{g\rho} - \frac{V\, dV}{g}$$

From the last two equations, therefore,

$$\frac{dp}{g\rho} + \frac{V\, dV}{g} + dF = 0 \qquad (43)$$

Equation (43) is the mechanical-energy-balance form of Bernoulli's theorem.

Under most circumstances, the exact integration of equation (43) is not possible. Consider, for example, the equation

$$\int_1^2 \frac{dp}{g\rho} + \frac{V_2^2 - V_1^2}{2g} + F = 0 \qquad (44)$$

The quantity ρ generally is an unknown function of p and the first term in equation (44) cannot be evaluated. The integration can be carried out for some special cases. For incompressible fluids, for example, ρ is constant. Also, for the isothermal flow of compressible fluids, ρ is proportional to p.

In the general case of compressible fluids, an approximate solution is

$$\frac{p_2 - p_1}{g\bar{\rho}} + \frac{V_2^2 - V_1^2}{2g} + F = 0 \qquad (45)$$

For passages of constant cross-sectional area,

$$\rho V = \text{Constant}$$

and equation (43) can be multiplied through by $g\rho$ and integrated to give

$$p_2 - p_1 + \rho V (V_2 - V_1) + \Delta p_f = 0 \qquad (46)$$

Equation (46) can be written

$$p_1 - p_2 = \Delta p = \Delta p_m + \Delta p_f$$

The equation is exact but, in general, Δp_f cannot be calculated exactly because the variation of V with L in the following equation is not known:

$$\Delta p_f = \frac{2\rho V}{D} \int_0^L fV \, dL$$

An approximate integration is

$$\Delta p_f = \frac{2f\rho V \bar{V} L}{D} \qquad (47)$$

The value of $\bar{V}$ may be taken as $\frac{V_1 + V_2}{2}$.

Adiabatic-Flow Equations

Equation (40) can be applied to the flow between stations 0 and 1 of figure 17. This flow is adiabatic, that is,

$$H = 0$$

Equation (40) then becomes

$$T_1 = T_0 + \frac{V_0^2 - V_1^2}{2g \, c_p \, J} \qquad (48)$$

This relation is used to calculate the temperature at station 1. For air,

$$T_1 = T_0 + \frac{0.832}{10^4} (V_0^2 - V_1^2) \qquad (49)$$

If losses in total pressure that may occur between stations 0 and 1 are neglected, the adiabatic relation

$$\frac{p}{\rho^{\gamma}} = \text{Constant}$$

holds, and it can be shown that the pressure and the density at station 1 are given by

$$\frac{T_1}{T_0} = \left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{\rho_1}{\rho_0}\right)^{\gamma-1} \tag{50}$$

Expressions for Total Pressure

The total pressure in a stream is defined as the static pressure that would exist if the stream were brought to rest, that is, the static pressure at a stagnation point. It can be shown (reference 10) that, in a compressible fluid, the total pressure is given by

$$H = p\left(1 + \frac{\gamma - 1}{2\gamma}\,\frac{\rho V^2}{p}\right)^{\frac{\gamma}{\gamma-1}} \tag{51}$$

For air, equation (51) becomes

$$\begin{aligned} H &= p\left(1 + 0.2\,\frac{\rho V^2}{\gamma p}\right)^{3.5} \\ &= p\left[1 + 0.2\left(\frac{V}{c}\right)^2\right]^{3.5} \end{aligned} \tag{52}$$

where the velocity of sound $c = \sqrt{\frac{\gamma p}{\rho}}$.

POWER COST

An approximate calculation of the power cost of a heat exchanger can be made by rather simple approximation formulas. The net power cost can be considered the sum of the power used in forcing the cooling air through the exchanger and the power used in transporting the weight of the exchanger minus the thrust power that is recovered from the heated cooling air. Simple approximation formulas can be used to calculate the cooling-air pumping power and the thrust power.

An exact determination of the power cost of a heat exchanger can be made by adding the cost of transporting the weight of the exchanger and the power expended in changing the momentum of the cooling air; the power expended in changing the momentum of the cooling air is

$$P = \frac{W}{g} V_0 (V_0 - V_3) \tag{53}$$

A rather complicated but exact method of finding V_3 is explained later.

The form drag of a heat-exchanger installation is not considered in the present discussion.

As in the preceding section, the formulas that are given herein for calculation of power cost are stated in terms of the conditions that exist at the stations 0 to 3 of figure 17.

Approximate Calculation of Power Cost

Cooling-air pumping power. - The power expended in pumping the cooling air between stations 1 and 2 is

$$P_c = \frac{W}{g}\left(\frac{p_1}{\rho_1} - \frac{p_2}{\rho_2}\right) \tag{54}$$

Although this expression is exact, it is a little too unwieldy for general use. It is more convenient to use the approximate expression

$$\begin{aligned} P_c &\approx \overline{Q} (p_1 - p_2) \\ &\approx \overline{Q}\, \Delta p \end{aligned} \tag{55}$$

where $\overline{Q}$ is the average volume rate of flow. The value of $\overline{Q}$ can be calculated from the average density or the average velocity

$$\overline{Q} \approx \frac{W}{g\overline{\rho}} \approx A\overline{V}$$

Weight-carrying power. - The power expenditure required to transport the weight of a heat exchanger is

$$P_W = \underbrace{\epsilon}_{\text{Mounting weight factor approx 1.5}} \frac{C_D}{C_L} \underbrace{\rho_R v}_{\text{H.X. Weight}} V_0 \tag{56}$$

The product $\rho_R v$ is the weight of the exchanger. The quantity ϵ is a factor that takes account of the weight of the mounting required by the exchanger. Experience has shown that ϵ generally has the approximate value 1.5.

Scoop-drag power. - For some installations, as when an exchanger is wholly or partly installed in an external scoop or duct, it may be desirable to take into account the drag that is associated with the frontal area of the scoop. The drag power of a scoop is given by

$$P_s = C_{D_s} q_0 V_0 A_s$$

If the scoop is well designed (see section entitled "The Cooling-Air Entrance"), the drag coefficient C_{D_s} based on the projected frontal area has a value of about 0.06. (See fig. 41 and reference 11.)

Conversion of heat energy into thrust power. - A part of the heat energy that is transferred to the air between stations 1 and 2 (fig. 17) can be converted into thrust power. Formulas for the approximate calculation of the recovered thrust power are given in the present section.

One of the thermodynamic processes by which heat energy is transformed into mechanical energy is known as the Joule cycle. In traversing this cycle, the working substance goes through the following stages: adiabatic compression; addition of heat at constant pressure; adiabatic isentropic expansion; return, at constant pressure, to initial conditions. The cycle is represented by the path ABCDA in figure 18. A somewhat similar cycle is traversed by the cooling air that flows through the arrangement depicted in figure 17. There is an adiabatic compression of the air between stations 0 and 1. Heat is transferred to the air between stations 1 and 2. The air expands adiabatically between stations 2 and 3. After passing station 3, the air returns to original free-stream conditions of temperature and density.

The rate W at which mechanical energy is recoverable from the heat input in the ideal cycle ABCDA is

$$W = JH \left[1 - \left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$= JH \left(1 - \frac{T_0 + 460}{T_1 + 460}\right) \qquad (57)$$

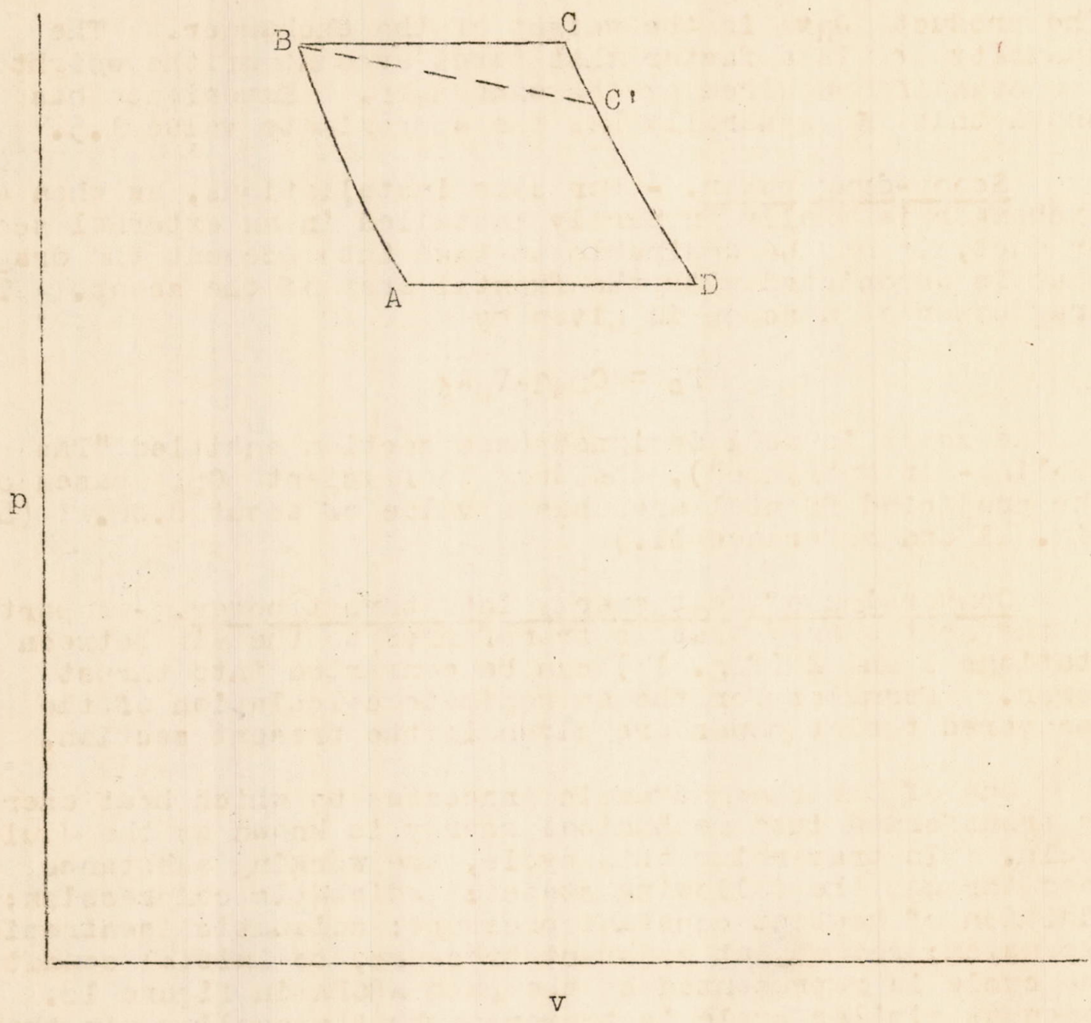


Figure 18. - Diagram of p against v for Joule cycle.

Three assumptions that were used in the derivation of equation (57), which are not exactly true in practice, cause the rate of recoverable mechanical energy in the actual cycle to be less than in the ideal cycle. One of these assumptions is that H is given by equation (4). Actually, equation (40) should be used. Another assumption is that the heat is transferred to the air at constant pressure. Actually, there is a fall in pressure between stations 1 and 2 and the actual path traversed during the cycle is somewhat as shown by ABC'DA in figure 18. A more nearly accurate value of W can be obtained by using in equation (57) the average of p_1 and p_2, instead of p_1. The third assumption is that the expansion between stations 2 and 3 is isentropic. In the actual case, this expansion entails a loss in total pressure.

In order to obtain an expression for the thrust power that is obtained from the heat input, the quantity W (equation (57)) must be multiplied by the efficiency of conversion of mechanical energy into thrust energy. This efficiency is $\frac{V_0 + V_3}{2V_0}$ if V_3 is less than V_0 and $\frac{2V_0}{V_0 + V_3}$ if V_3 is greater than V_0.

According to equation (57), the efficiency of the Joule cycle W/JH depends on ΔT, the rise in air temperature between the free stream and the heat-exchanger entrance, because

$$T_1 = T_0 + \Delta T$$

By equation (49), ΔT is proportional to the difference of the squares of the free-stream velocity and the velocity in the heat-exchanger entrance and

$$\Delta T = \frac{0.832}{10^4} (V_0^2 - V_1^2)$$

The efficiency of the cycle accordingly is quite small unless V_0 is large (of the order of 600 fps or more) and V_1 is much smaller. At low altitudes where V_0 is small, therefore, the mechanical power obtained by conversion of the heat input is quite small.

It might be expected that the efficiency of the cycle is highest at highest altitudes where V_0 is largest (if it is assumed that the critical altitude is not exceeded). That this fact is not necessarily true can be shown as follows: Consider the isothermal region of the atmosphere in which T_0 is constant. At first, as the altitude of flight is increased, V_0, the quantity $V_0^2 - V_1^2$, and the efficiency W/JH increase. As the altitude increases further, however, the heat exchanger may become unable to perform the necessary heat transfer unless V_1 increases rapidly. In that case, although V_0 increases, $V_0^2 - V_1^2$ and the efficiency decrease. These effects are shown in figure 19, which is taken from figure 20 of reference 12.

Exact Calculation of Power Cost

If the velocity of the cooling air at the exit from the duct V_3 is known, the net cost of cooling (exclusive of the weight-carrying power) can be calculated by the

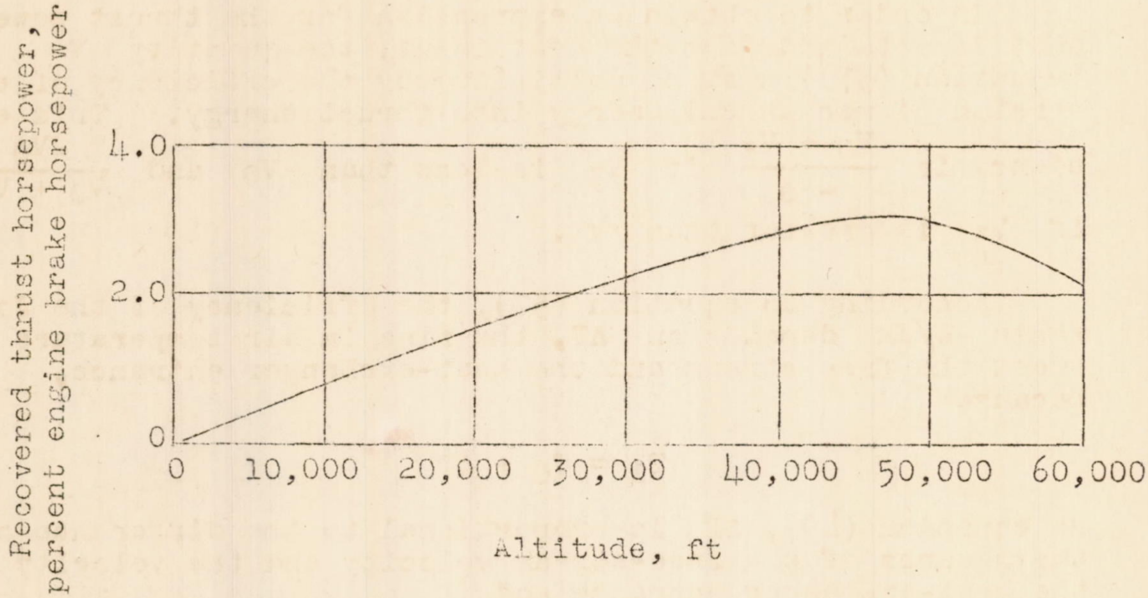


Figure 19. - Thrust horsepower recovered from cooling air in percent of engine brake horsepower as a function of altitude for the airplane considered in reference 12. (From fig. 20 of reference 12.)

equation

$$P = \frac{W}{g} V_0 (V_0 - V_3) \tag{53}$$

Two methods of finding V_3 are given here. The first consists of station-to-station calculations. The second can be used when the loss in total pressure across the entire system, that is, between stations 0 and 3, is known.

By the first method, the value of V_3 is obtained by first calculating the conditions at station 1 from those at station 0, then the conditions at station 2 from those at station 1, and finally the conditions at station 3 from those at station 2. The equations that are needed in the calculations have already been given in the section entitled "Energy Balances." The method of calculation is illustrated by the following example (see also reference 13), which is simplified by assuming values for certain quantities.

Assume that the heat exchanger (fig. 17) has an open frontal area A_1 of 1.8 square feet, that it dissipates heat at the rate H of 353 Btu per second, that it requires a weight rate of flow W of 14.34 pounds per second, and that

the friction pressure drop Δp_f in the exchanger is 36 pounds per square foot. Assume that the airplane is flying at 600 feet per second at an altitude of 25,000 feet in NACA standard atmosphere.

Conditions at station 1. - The value of V_1 can be found by a simultaneous solution of equations (49), (50), and the equation

$$W = gA\rho V$$

The values of T_1, p_1, and ρ_1 are obtained from equations (49) and (50). The values are given in table IV.

TABLE IV. - VALUES OF QUANTITIES USED IN EXAMPLE

Station	V (fps)	ρ (slug/cu ft)	p (lb/sq ft)	T (°F abs.)	A (sq ft)
0	600	0.001065	785	429	----
1	200	.001238	966	455	1.8
2	258	.000960	916	556	1.8
3	596	.000860	785	532	.87

Conditions at station 2. - The values of the variables at station 2 can be obtained by a simultaneous solution of equations (42) and (46) and the equation of continuity

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \tag{58}$$

in which $A_1 = A_2$. The solution of these equations gives, for the conditions at station 2, the values shown in table IV.

Conditions at station 3. - The static pressure at station 3 is very nearly the static pressure that exists outside the duct in the region of station 3. If flaps are not used at the exit, the pressure there is practically free-stream pressure. If p_0 is substituted for p_3, the exit velocity, temperature, and density can be calculated from the equations for adiabatic changes (equations (49) and (50))

$$T_3 = T_2 + \frac{0.832}{10^4}\left(V_2^2 - V_3^2\right) \tag{49}$$

and

$$\frac{T_3}{T_2} = \left(\frac{p_0}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{\rho_3}{\rho_2}\right)^{\gamma-1} \tag{50}$$

The values are as given in table IV. As was pointed out in reference 13, slide-rule calculations are not sufficiently accurate for a satisfactory determination of V_3 from the two preceding equations.

As soon as the value of V_3 is known, the net power cost of the heat exchanger (exclusive of the weight-carrying power) is easily obtained from

$$P = \frac{W}{g} V_0 (V_0 - V_3) \tag{53}$$

$$= \frac{14.34}{32.2} \times 600 \times (600 - 596)$$

$$= 1070 \text{ ft-lb/sec}$$

Simplified calculation of power cost. - A single equation can be developed that will give the power cost of the heat exchanger, provided that the loss in total pressure between stations 0 and 3 is known. Equation (41), the energy-balance equation, can be applied between stations 0 and 3 to give

$$\frac{V_3^2}{2} + \frac{\gamma}{\gamma - 1} \frac{p_3}{\rho_3} = \frac{V_0^2}{2} + \frac{\gamma}{\gamma - 1} \frac{p_0}{\rho_0} + Jg \frac{H}{W} \tag{59}$$

The ratio p_3/ρ_3 can be eliminated as follows: The total pressure at station 3 is equal to the stagnation pressure that would exist if the fluid were brought to rest and is given by equation (51) as

$$H_3 = p_3 \left(1 + \frac{\gamma - 1}{2\gamma} \frac{\rho_3 V_3^2}{p_3}\right)^{\frac{\gamma}{\gamma-1}} \tag{51}$$

Then,

$$\frac{p_3}{\rho_3} = \frac{\gamma - 1}{2\gamma} \frac{V_3^2}{\left(\frac{H_3}{p_3}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

By substituting this expression for p_3/ρ_3 in equation (59) and simplifying, an expression for V_3 is obtained in the

form

$$\frac{V_3{}^2}{2} = \left(\frac{V_0{}^2}{2} + \frac{\gamma}{\gamma - 1}\frac{p_0}{\rho_0} + Jg\frac{H}{W}\right)\left[1 - \left(\frac{p_3}{H_3}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad (60)$$

The equation is easily modified to read

$$\frac{V_0 - V_3}{V_0} = 1 - \left(1 + \frac{1}{0.2M_0{}^2} + \frac{2\ JgH}{V_0{}^2\ W}\right)^{\frac{1}{2}}\left[1 - \left(\frac{p_3}{H_3}\right)^{\frac{1}{3.5}}\right]^{\frac{1}{2}} \qquad (61)$$

For p_3 there can be used $k_p p_0$, where

$$k_p = \frac{p_3}{p_0}$$

If flaps are not used at the exit, the pressure coefficient k_p has a value very close to unity. If the loss in total pressure ΔH is known, H_3 can be found as

$$H_3 = H_0 - \Delta H$$

and equation (61) can be used with equation (53)

$$P = \frac{W}{g} V_0 (V_0 - V_3) \qquad (53)$$

to find the power cost.

APPLICATIONS TO DESIGN

The information necessary for calculating rate of heat transfer, pressure drop, and power cost for various designs has been presented. A number of questions arising from application of this information to the design of heat exchangers remain to be discussed.

Size of Passages

The pressure drop available for a heat exchanger is frequently none too large, especially at high altitudes. The pressure drop required by a heat exchanger can be reduced, for a given rate of heat transfer, by increasing the amount of surface area S and decreasing the velocity V and the heat-transfer coefficient h. Pressure drop is proportional

to $V^{1.8}$. The quantity H is proportional to $SV^{0.8}$. If, therefore, V is reduced and S is increased sufficiently for the product $SV^{0.8}$ to remain constant, a relatively large reduction in $V^{1.8}$ and in pressure drop follows. Furthermore, the installation space available on airplanes for heat exchangers is usually rather limited and difficulty in fitting a heat exchanger into the available space is frequently experienced. A good design of heat exchanger consequently is one in which much heat-transfer surface area is crowded into a small volume. Fluid passages should be made as small as possible, provided that the requirements of mechanical strength can be met and flow in the turbulent or the transition region can be obtained.

Tube Spacing and Arrangement

A good method of comparing various designs is by means of plots of heat-transfer coefficient against pumping power cost per unit heat-transfer surface area. (See reference 14.) Figures 29 and 30 of reference 14, which are based on the tests reported in reference 15, are reproduced herein with some slight modifications as figures 20 to 23. In figures 20 to 23, heat-transfer coefficient h is shown as a function of pumping power cost per unit surface area times the square of the density $P_c\rho^2/S$ for flow across staggered and in-line tube banks. These figures show clearly that the smallest practicable tube spacing both in the direction parallel to the air flow and in the direction perpendicular to the air flow results in the smallest power cost for a given value of heat-transfer coefficient. This result is true for both the staggered and the in-line tubes. Figures 20 to 23 also show that, for given values of the spacing factors and for a given value of heat-transfer coefficient, the in-line arrangement results in slightly lower values of power cost than the staggered arrangement.

Comparison of Flow across Tubes with Flow through Tubes

A comparison of the plots of heat-transfer coefficient against power cost per unit surface area (figs. 20 to 23) for flow across banks of circular tubes (solid lines) and for flow inside smooth tubes (broken lines) shows that flow across tubes is more economical than flow through tubes; that is, for a given value of heat-transfer coefficient, the pumping power is less for flow across tube banks than for flow through tubes, except at extremely high values of Reynolds number. In the plots for flow through tubes, two

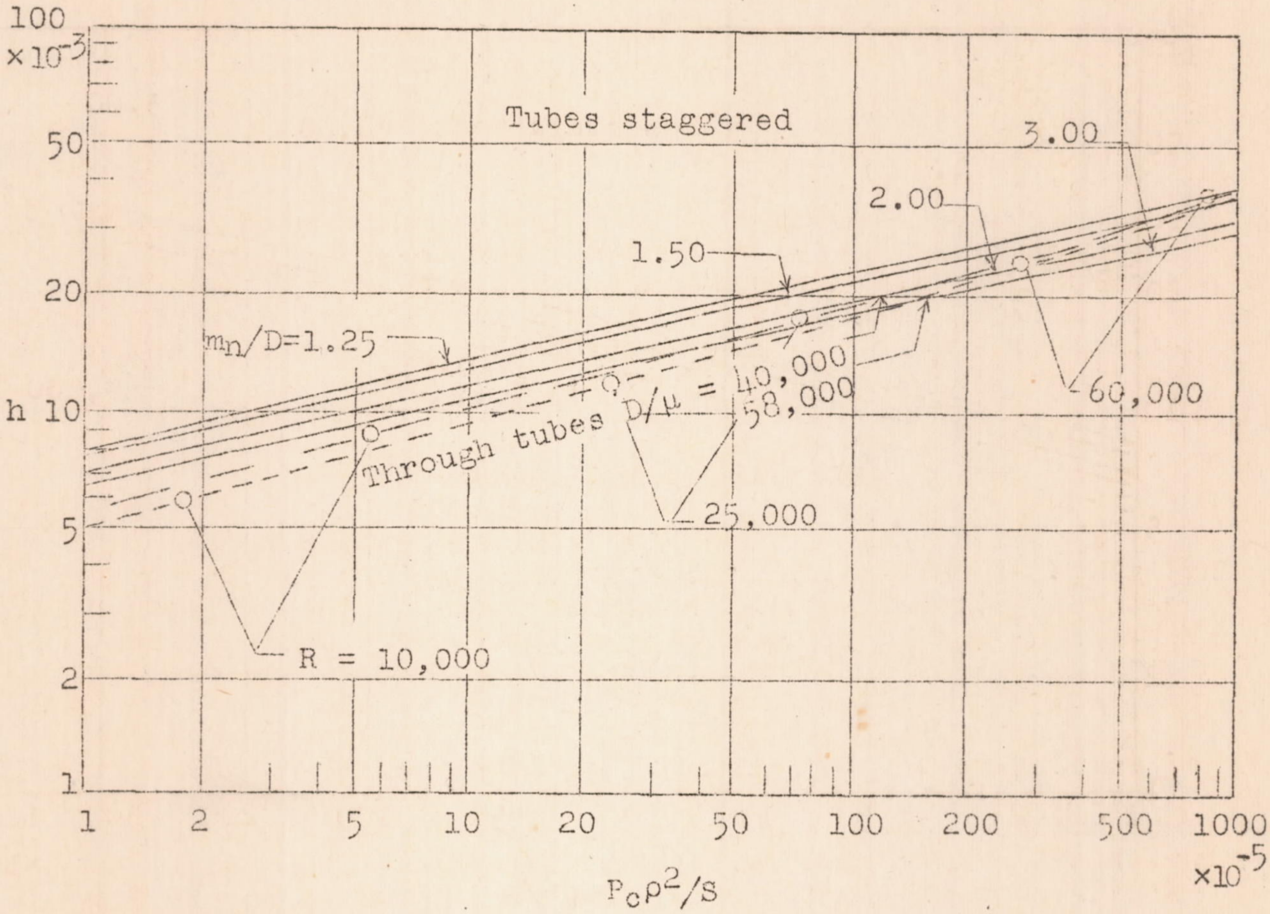


Figure 20. - Efficiency of heat transfer - heat-transfer coefficient h as a function of spacing factor m_n/D and of pumping power per unit surface area times the square of the density. Flow of air across staggered tubes and flow through tubes. (From fig. 30 of reference 14.)

values of the parameter D/μ are used. The value of D/μ of 58,000 corresponds to a tube diameter of 0.26 inch for the standard sea-level coefficient of viscosity μ of 3.73×10^{-7} slug per foot per second. The value of D/μ of 40,000 corresponds, for the same value of μ, to a tube diameter of 0.18 inch.

Choice of Fluid-Surface Combination

If the two surfaces of a heat exchanger differ in arrangement or amount (as, for example, when one fluid flows through the tubes and the other flows across the tubes or

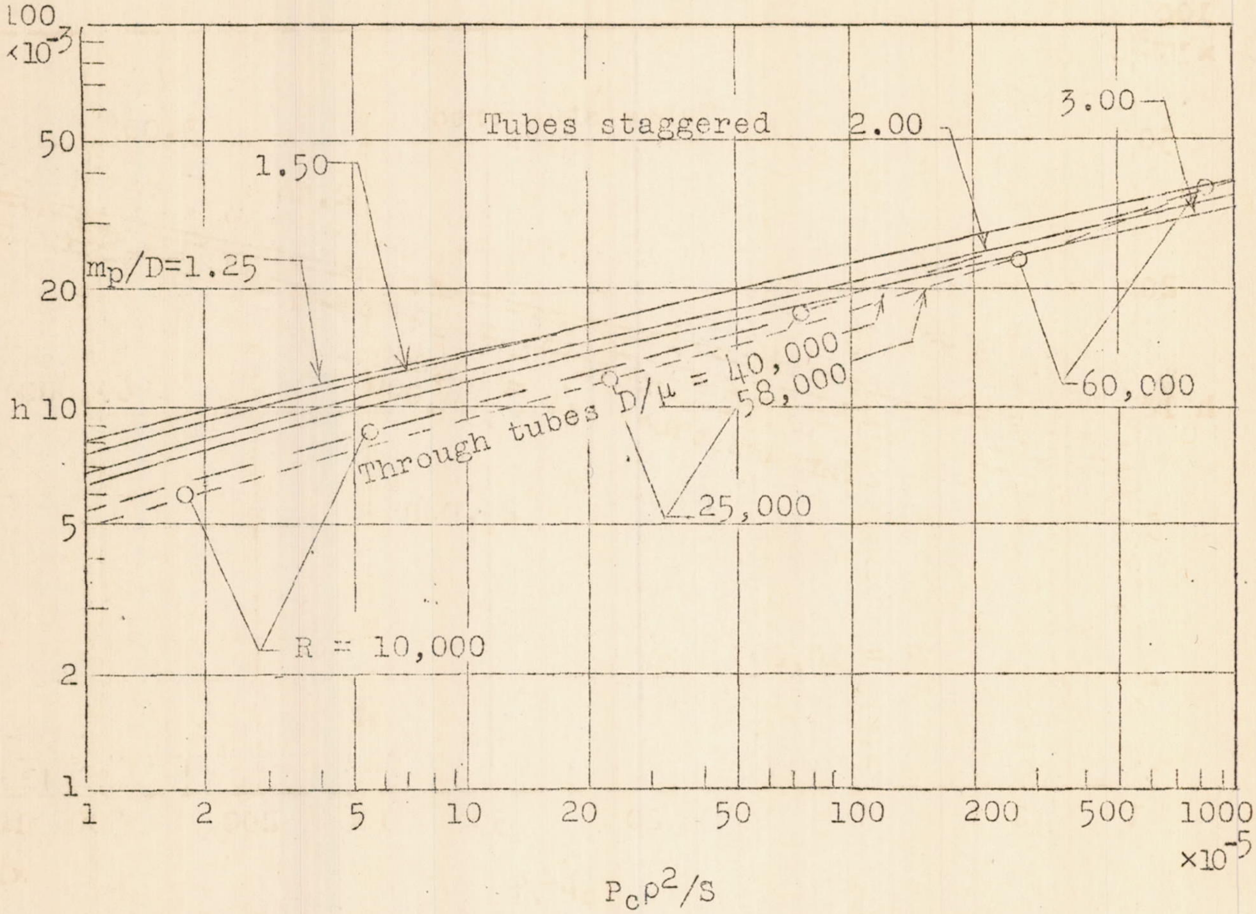


Figure 21. - Efficiency of heat transfer - heat-transfer coefficient h as a function of spacing factor m_p/D and of pumping power per unit surface area times the square of the density. Flow of air across staggered tubes and flow through tubes. (From fig. 29 of reference 14.)

when only one surface is finned), the question of which fluid should flow across which surface arises. For given values of S_1 and S_2, the over-all thermal resistance $1/h_tS_t$, which is given by

$$\frac{1}{h_tS_t} = \frac{1}{h_1S_1} + \frac{1}{h_2S_2} \qquad (23)$$

is a minimum when the thermal resistances on both sides are equal. From considerations of economical heat transfer, therefore, the question of which fluid should go with which surface is answered by so making the choice that h_1S_1 is more nearly equal to h_2S_2.

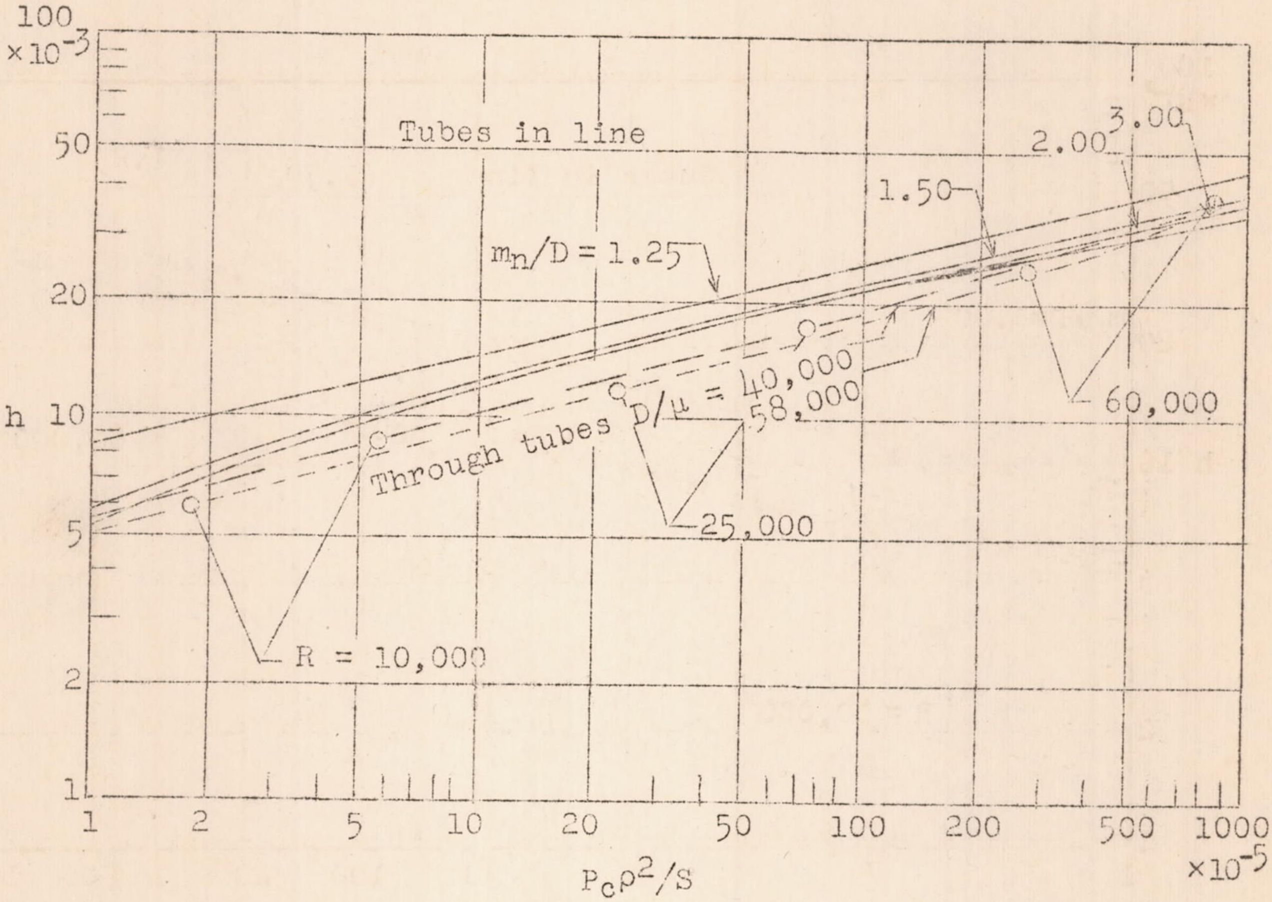


Figure 22. - Efficiency of heat transfer - heat-transfer coefficient h as a function of spacing factor m_n/D and of pumping power per unit surface area times the square of the density. Flow of air across in-line tubes and flow through tubes. (From fig. 30 of reference 14.)

From mechanical considerations, the tube-wall thickness can be smaller and the shell can be lighter, and the weight of the exchanger consequently can be less, if the high-pressure fluid is inside and the low-pressure fluid is outside the tubes.

Comparison of Counterflow with Crossflow

As has been pointed out for the same conditions of weight rate of flow, the over-all mean temperature difference ζ is greater for counterflow than for crossflow. The counterflow intercooler has often been considered for use in airplanes. In the following discussion, however, it is shown that the counterflow arrangement is not so advantageous as is frequently believed.

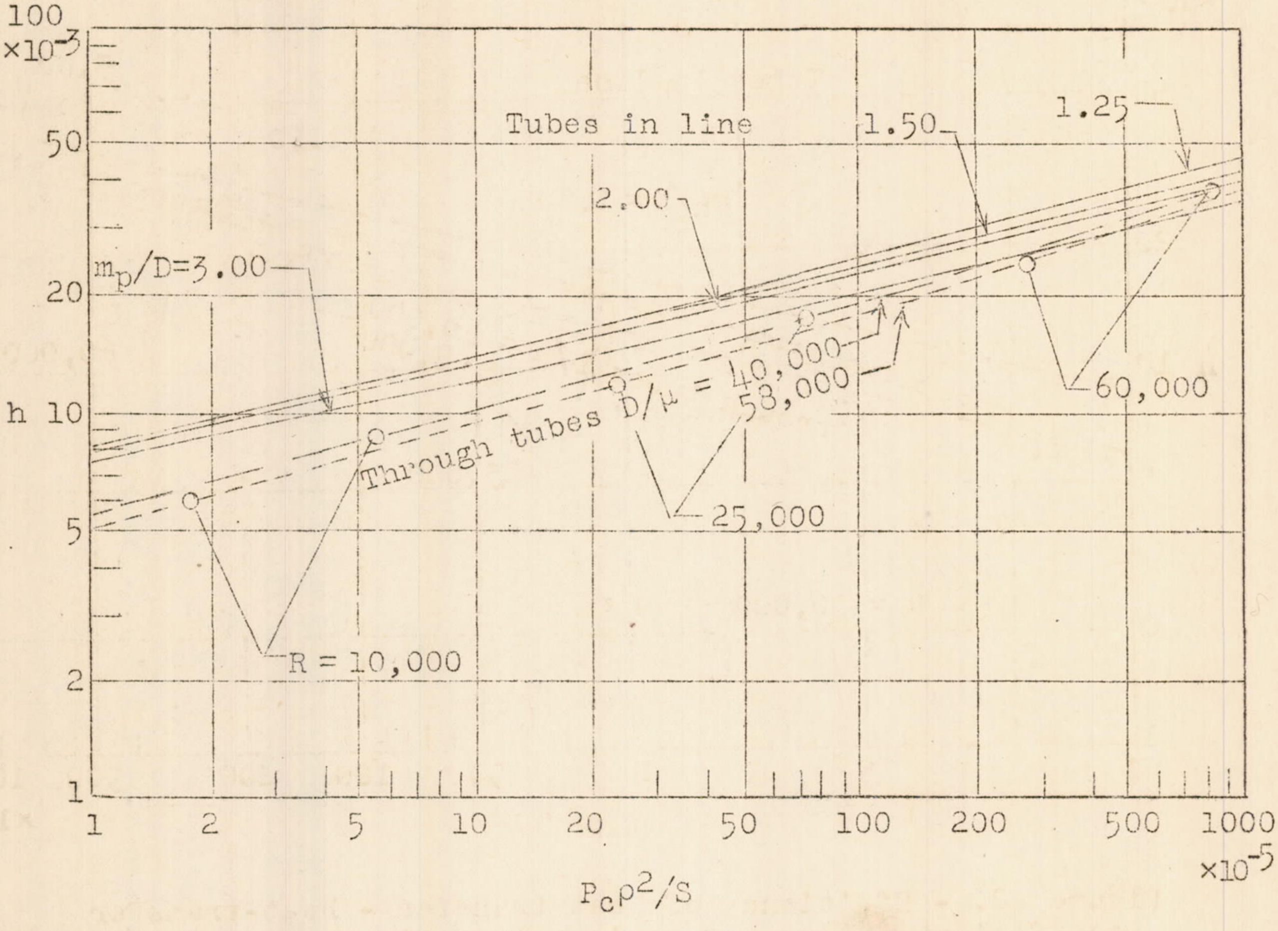


Figure 23. - Efficiency of heat transfer - heat-transfer coefficient h as a function of spacing factor m_p/D and of pumping power per unit surface area times the square of the density. Flow of air across in-line tubes and flow through tubes. (From fig. 29 of reference 14.)

Table III shows the values of the ratio of ζ for crossflow to ζ for counterflow for various values of the hot-fluid temperature drop ξ and the cold-fluid temperature rise η. The value of the ratio is quite low for high values of ξ and η. Consider, however, the case of high-altitude intercoolers. At high altitudes where much supercharging is necessary, ξ is large. On the other hand, for the efficient operation of high-altitude intercoolers, the ratio W_c/W_e must be relatively large in order for ζ to be large and η, as given by the equation

$$W_e \xi = W_c \eta$$

must therefore be small. Inspection of table III shows that, for large ξ and small η, the value of ζ for counterflow is only a small percentage greater than the value of ζ for crossflow. It would be a mistake to design a counterflow intercooler for large ξ and large η in order to take advantage of the large difference that would then exist between the counterflow ζ and the crossflow ζ. For large ξ and large η, the value of ζ for both counterflow and crossflow is much smaller than its value for large ξ and small η and more surface area would be required for the small ζ for a given rate of heat dissipation H. These facts are illustrated by figure 24, which

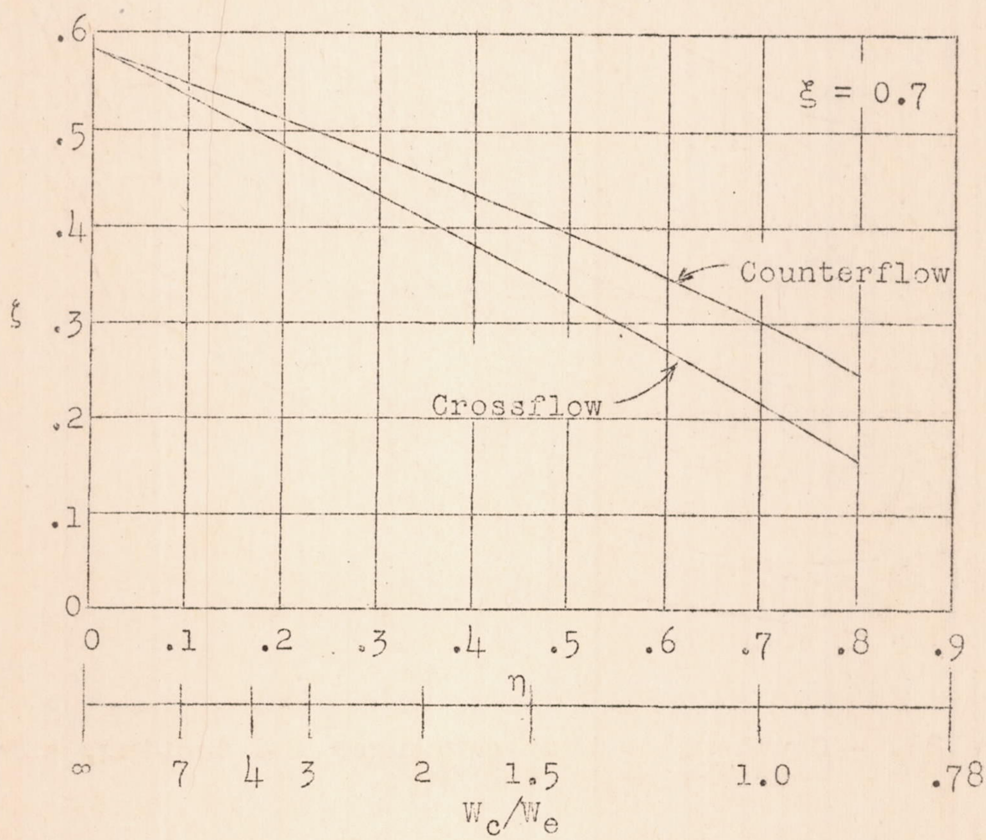


Figure 24. - Comparison of mean temperature difference in counterflow and crossflow. ζ as a function of η; $\xi = 0.7$.

is drawn for a value of ξ of 0.7, which is necessary at an altitude of about 35,000 feet. Figure 24 shows that, at large values of η, counterflow gives a large percentage increase over crossflow in the over-all temperature difference ζ. For small values of η, however, the percentage increase is small.

Another factor to be considered is that the manifolding necessary with a counterflow unit (see fig. 25) must contain much heat-transfer surface area and that the flow across this surface area is very nearly crossflow.

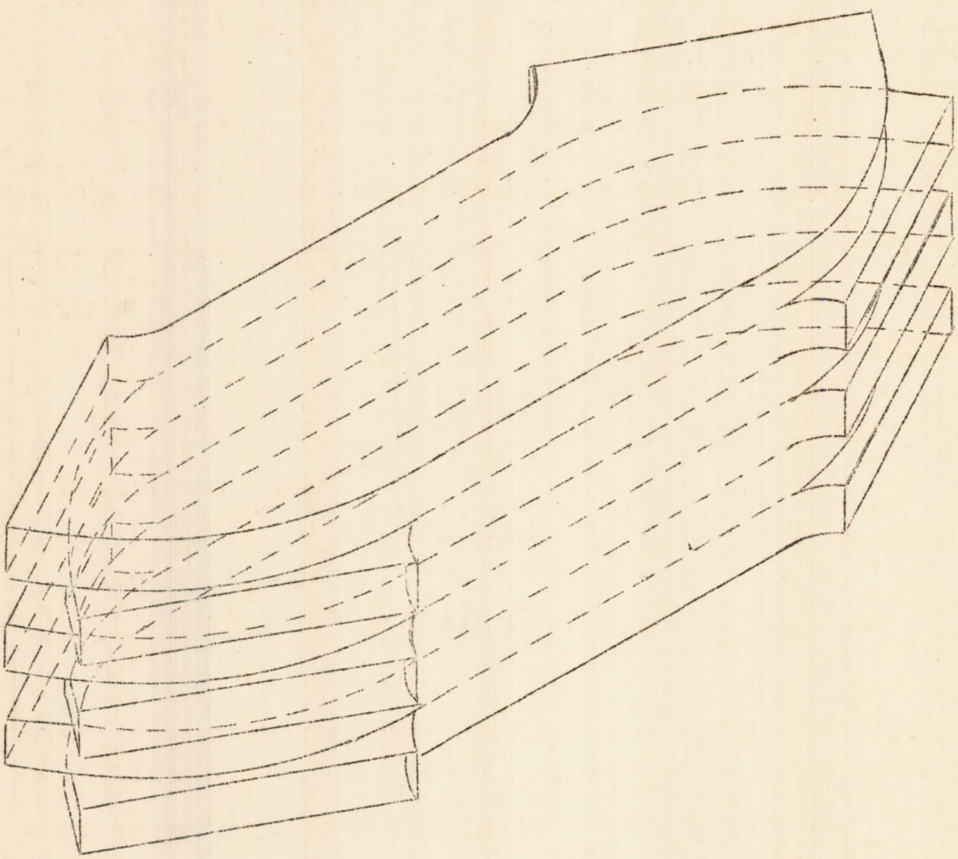


Figure 25. - Counterflow heat exchanger and ducting, showing crossflow in ducting.

Mechanical Considerations

One of the most important problems connected with the development of heat-transfer apparatus is the economical production of exchangers that are free from the tendency toward structural and leakage failures. Vibration usually imposes a severe strain on heat exchangers. The differences in the efficiency of heat transfer for different designs is small, and these differences are negligible if there are important differences in mechanical reliability. There have

been innumerable attempts, nevertheless, to obtain large increases in efficiency by the use of new tube shapes and novel assembly arrangements. Unless these new shapes and arrangements result in smaller tube diameters, more substantial construction, or more economical manufacture, they can have little advantage over the shapes and arrangements in current use.

It would be desirable in many instances to use rather long tubes. Long tubes, because of vibration, must be supported. The use of long tubes introduces a manufacturing hardship, however, that tends to keep heat-exchanger tubes short. Vibration difficulties are more serious when the tubes are filled with liquid than when the liquid is outside the tubes and the cooling air inside.

For coolant radiators, hexagonal tubes would be desirable on the basis of ratio of open to total frontal area but are probably unsuitable because of tendency to failure under vibration.

Oil coolers of the proper shape (rectangular) to fit the available space on an airplane would be desirable from installation considerations. Such coolers, however, would have to be excessively heavy to avoid failure under high oil pressure. Oil coolers therefore are usually of circular cross section.

In spite of the well-recognized advantages of small tube diameter, tubes smaller than 0.25 inch are rarely used because of the problem of assembly and because of the problem of reducing the tube-wall thickness in proportion to the reduction in diameter.

Oil Coolers

The design of oil coolers is at present well standardized with respect to dimensions. Circular coolers are produced in a number of diameters. Tube length, diameter, and spacing are limited to a few values. The cooling air flows through the tubes and the oil flows across the tubes. Baffles are used in the oil passages. The different baffles used by different manufacturers result in somewhat different heat-transfer performances, particularly at low cooling-air inlet temperatures for which the influence of baffle arrangement on the tendency of the oil to congeal is particularly strong.

At present, the principal problem connected with oil coolers is the problem of congealing. Oil is so highly viscous, compared with the fluids used in radiators and intercoolers, that the oil flow is probably always laminar, whereas the flow of glycol and air is almost always turbulent. When the inlet temperature of the cooling air is low, the laminae of oil next to the cooling surface become cooled to such an extent that their coefficient of viscosity rises to a much higher value than that of laminae farther from the surface. The velocity of the cold oil is reduced, and the cold oil remains longer in contact with the tube surface and probably assumes a temperature close to that of the tube wall. The result is that the passageways eventually become clogged with cold or congealed oil and the cooler ceases to function. The design of oil coolers that are not subject to the tendency of the oil to congeal at high altitudes (low cooling-air temperatures) is a problem that has not yet been solved.

Because the flow of oil in coolers is laminar and because of the various effects of the baffle arrangements used by different manufacturers, little progress has been made in correlating the heat transfer on the oil side.

II - S E L E C T I O N

Part I has dealt principally with the internal design of heat exchangers and the effect of internal design on rate of heat transfer, pressure drop, and power cost. The next problem that confronts a heat-transfer engineer is the question of selection; that is, after the internal geometry of a heat exchanger has been chosen, what external dimensions and what cooling-air flow rate should be selected? These quantities should be so selected

(1) that the heat exchanger will be capable of the required rate of heat dissipation

(2) that the cooling-air pressure drop necessary to effect the required heat dissipation will not exceed the amount available

(3) that none of the dimensions will exceed the limits set by the installation space available on the airplane

(4) that the power consumption of the heat exchanger will not be excessively greater than the lowest power consumption attainable by use of dimensions of the proper proportions

(5) that the allowable engine-air pressure drop (if the exchanger is an intercooler) is not exceeded

It is generally possible to calculate, from the equations of part I, the solution to the problem of selection. The calculations are unusually tedious, however, particularly in view of the fact that a number of trials is always necessary to find a heat exchanger that satisfies all the requirements. A very much more satisfactory method of selection is by a chart. Two general types of chart have been devised that might be termed performance charts and selection charts. The present section is a consideration of such charts and their use in selecting coolant radiators, air intercoolers, and oil coolers.

SELECTION OF RADIATORS

Most coolant radiators are so constructed that the cooling air flows through smooth tubes and the coolant flows across and around the tubes. The equations that describe

the performance of such radiators have already been discussed and are restated hereinafter (equations (62), (19), and (31)). The problem of selection of radiators consists in finding values for the five variables P_t, Q, Δp, v, and L that satisfy these three equations and the requirements just listed. This problem may be solved conveniently by use of a generalized selection chart that shows the relations among the variables for any given set of operating conditions. Such a selection chart is presented in reference 16. A résumé of the derivation of the chart and an example of its use are given in the following paragraphs.

The power cost of a radiator is

$$P_t = Q\ \Delta p + \epsilon \frac{C_D}{C_L} \rho_R V_0 v \tag{62}$$

(See equations (55) and (56).) The rate of heat dissipation is

$$H = W c_p \Delta T_i \left(1 - e^{-0.1R^{-0.2} \frac{L}{D}}\right) \tag{19}$$

The pressure drop is

$$\Delta p = 0.092\ \rho V^2 \frac{L}{D} R^{-0.2} \tag{31}$$

From equations (62), (19), and (31) and the equation

$$L = \frac{v}{A}$$

where v is the open volume of the heat exchanger and A is the open frontal area of the heat exchanger, the following relations can be derived:

$$\Delta p' + \left(\frac{P'}{v'} - 1\right)^{\frac{5}{7}} \log_e \left(1 - \frac{\Delta p'}{P' - v'}\right) = 0 \tag{63}$$

$$\Delta p' - \left(\frac{Q'}{v'}\right)^{\frac{5}{2}} \left(\log_e \frac{1}{1 - \frac{1}{Q'}}\right)^{\frac{7}{2}} = 0 \tag{64}$$

$$\Delta p' - \left[\frac{A'\left(\frac{A'\ \Delta p'}{v'}\right)^{\frac{5}{9}}}{v'}\right]^{\frac{5}{2}} \left\{-\log_e \left[1 - \frac{1}{A'\left(\frac{A'\ \Delta p'}{v'}\right)^{\frac{5}{9}}}\right]\right\}^{\frac{7}{2}} = 0 \tag{65}$$

The variables used in equations (63) to (65) are defined by the relations

$$v' = v\, K_1^{\frac{5}{7}} K_2$$

$$\Delta p' = \Delta p \frac{K_1^{\frac{5}{7}}}{\rho V_0^2}$$

$$p' = p_t \frac{K_1^{\frac{5}{7}} K_2}{C}$$

$$A' = A \frac{K_1^{\frac{9}{14}} K_2 R_0^{0.2} D}{4C_1}$$

$$Q' = Q \frac{K_1 K_2 R_0^{0.2} D}{4C_1 V_0}$$

where

$$C = \epsilon \frac{C_D}{C_L} \rho_R V_0$$

$$C_1 = 0.023$$

$$K_1 = \frac{4C_1 \rho V_0^3}{C\, D\, R^{0.2}}$$

$$K_2 = \frac{C\ c_p\ g\ (T_w - T_i)}{V_0^2\ H}$$

These primed variables, as related by equations (63) to (65), were plotted to obtain a selection chart (fig. 26, which is taken from fig. 1 of reference 16). This chart gives the general solutions of these equations. Each point on the chart represents a radiator. Because of the generalized form of the variables, the chart can be used for any set of operating conditions. All the quantities that are functions of the operating conditions, such as the characteristics and the performance of the airplane, the altitude of operation, the rate of heat dissipation, and the tube diameter, are taken care of in the constants in the definitions of the primed variables.

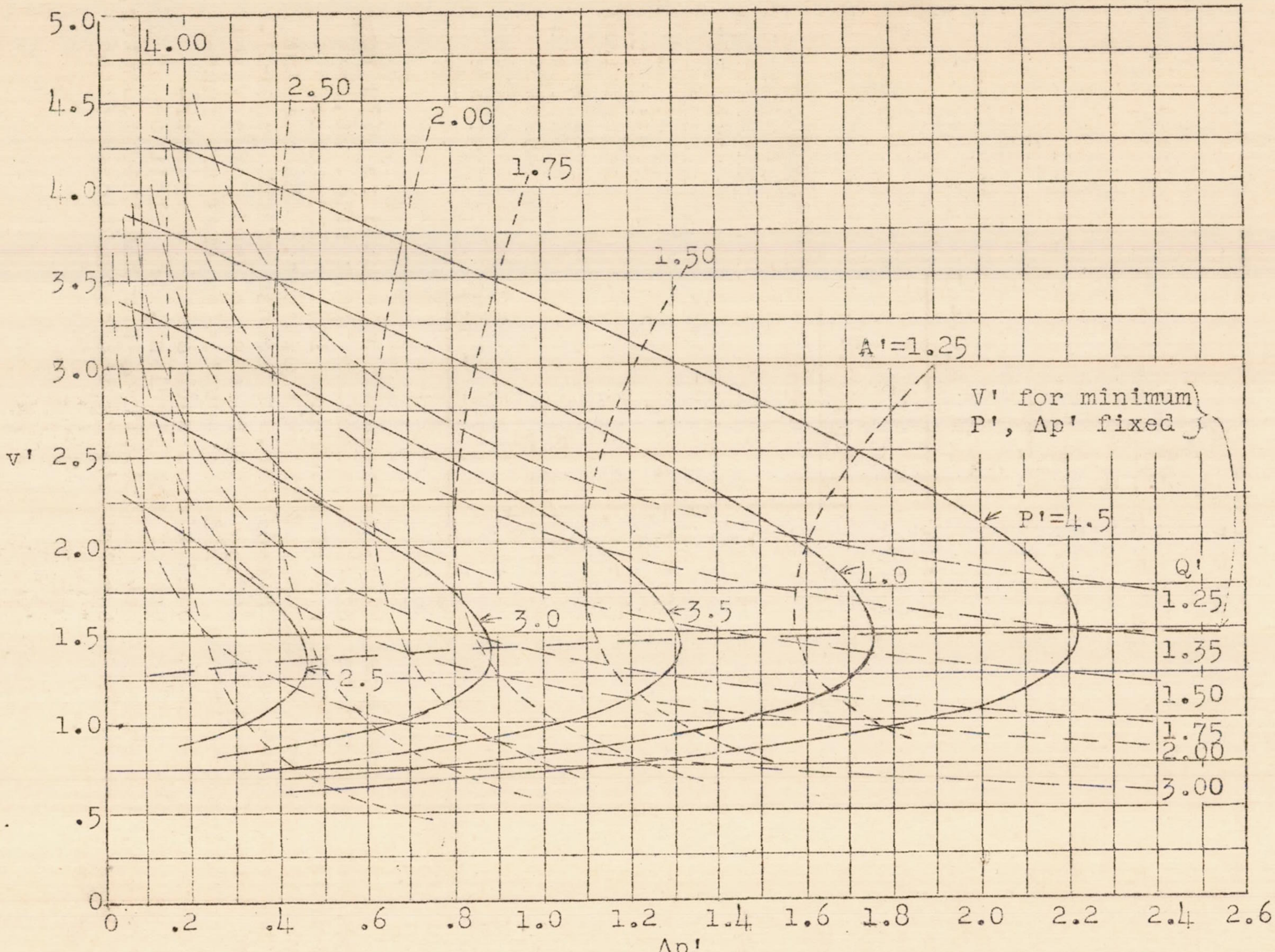


Figure 26. - Generalized radiator selection chart. (From fig. 1 of reference 16.)

The use of the chart is not difficult. One of the variables may be chosen in advance. (If two variables are chosen, the radiator is completely determined and all variables are fixed.) Either the pressure drop or the frontal area is usually selected in advance. For example, suppose the pressure drop Δp is chosen. Accordingly, $\Delta p'$ is a known fixed quantity. The point on the chart that represents the radiator having the given pressure drop may be chosen anywhere on the vertical line that has the proper value of $\Delta p'$. It is easy to locate the point on this line that represents the radiator of minimum power consumption or the point that represents the radiator of minimum frontal area. Correspondingly, a value may first be chosen for the frontal area A. Then any point on the corresponding A' contour may be selected. The radiator of minimum power may be found, or some other radiator that represents a better compromise among radiator volume, pressure drop, and volume rate of air flow may be chosen.

Before the use of the chart in selecting a radiator is illustrated, a remark on altitude is in order. The altitude for which the radiator should be chosen is one of the factors to be considered. As the altitude of flight increases, the decreasing density of the air tends to make cooling more difficult and the decreasing temperature of the air, up to the isothermal region, tends to make cooling less difficult. The most difficult cooling case ordinarily is at the critical altitude. In order to obtain maximum airplane speed, the radiator must be selected for the high-speed flight condition at the critical altitude. If the radiator is so chosen, however, the case of full-power climb at the critical altitude should be investigated to make sure that the pressure drop required by the radiator is not greater than the pressure drop available in climb.

In order to simplify and to systematize the calculations involved in selecting a radiator, selection forms have been devised for use in conjunction with the selection chart. The following example shows in detail how the forms and the chart are used. The procedure falls naturally into several steps, as follows:

(1) Fill in radiator selection form 1.

In filling in this form for the example, a typical set of operating conditions has been used.

Radiator Selection Form 1

Quantity	Symbol	Value	Unit
Heat dissipation	H	54.5	Btu/sec
Altitude		36,000	ft
Airplane speed	V_0	678	ft/sec
Airplane drag-lift ratio	C_D/C_L	0.09	
Weight factor	ϵ	1.5	
Specific weight of radiator	ρ_R	90	lb/cu ft of open volume
Radiator tube diameter	D	1/48	ft
Temperature of air at altitude	T_0	-67	°F
Estimated adiabatic temperature rise		34.9	°F
Inlet temperature of air	T_i	-32.1	°F
Density of air at altitude	ρ_0	0.000704	slug/cu ft
Estimated percentage of density increase		24	percent
Inlet density of air	ρ	0.000871	slug/cu ft
Coefficient of viscosity of air	μ	3.19×10^{-7}	slug/ft-sec
Radiator tube wall temperature = av. coolant temperature	T_W	290	°F
$\epsilon \frac{C_D}{C_L} \rho_R V_0$	C	8233	
$\frac{\rho V_0 D}{\mu}$	R_0	38,600	
	$R_0^{0.2}$	8.28	
$\frac{0.098 \rho V_0^3}{CD R_0^{0.2}}$	K_1	18.78	
	$K_1^{5/7}$	8.124	
	$K_1^{9/14}$	6.589	
$\frac{7.73 C(T_W - T_i)}{H V_0^2}$	K_2	0.08187	
Ratio of open to total frontal area of radiator	f_R	0.67	

(2) Using the data of form 1, fill in the column "Value" under "Constant" in form 2.

Radiator Selection Form 2

Constant		Generalized variable (a)		Variable		
Symbol	Value	Symbol	Value	Symbol	Value	Unit
$10.2K_1^{\frac{9}{14}}K_2R_0^{0.2}D$	0.946	A'	1.9	A	2.00	sq ft
$\dfrac{K_1^{\frac{5}{7}}}{\rho V_0^2}$	0.0203	Δp'	0.746	Δp	36.77	lb/sq ft
$\dfrac{10.2K_1K_2R_0^{0.2}D}{V_0}$	0.00398	Q'	1.94	Q	486.5	cu ft/sec
$K_1^{\frac{5}{7}}K_2$	0.665	v'	1.40	v	2.113	cu ft
$\dfrac{K_1^{\frac{5}{7}}K_2}{C}$	8.07×10^{-5}	P'	2.88	P_t	35,670	ft-lb/sec

[a]Constant × variable = generalized variable.

(3) Calculate the value of one of the generalized variables.

As stated before, one of the variables is usually chosen in advance. The present example is worked for a radiator that is to have a total frontal area of 3 square feet or an open frontal area A of $3f_R = 2$ square feet. Multiplication of this value of A by the value of the proper constant (see first line of selection form 2) gives 1.9 as the value of A'.

(4) Find the point on the selection chart that is to represent the radiator.

Any point on the line $A' = 1.9$ on the selection chart can be selected as the point representing the radiator to be selected. The values of the other radiator variables will depend upon the point selected. It seems reasonable to choose the point at which P' (and consequently P_t) is a minimum. This point will, in general, be near the minimum value of $\Delta p'$ (and of Δp). For $A' = 1.9$, the minimum P' is 2.88. At the point determined by these values of A' and P', $\Delta p' = 0.746$, $v' = 1.40$, and $Q' = 1.94$.

(5) Calculate the values of the radiator variables.

The values of the generalized variables and of the constants in selection form 2 are now known. From the relations shown in form 2, the values of Δp, Q, v, and P_t can be easily calculated:

$$\Delta p = \frac{\Delta p'}{0.0203}$$

$$= \frac{0.746}{0.0203}$$

$$= 36.8 \text{ lb/sq ft}$$

$$Q = \frac{Q'}{0.00398}$$

$$= \frac{1.94}{0.00398}$$

$$= 486 \text{ cu ft/sec}$$

$$\text{Open volume, } v = \frac{v'}{0.665}$$

$$= \frac{1.40}{0.665}$$

$$= 2.113 \text{ cu ft}$$

$$\text{Total volume} = \frac{v}{f_R}$$

$$= \frac{2.113}{0.67}$$

$$= 3.15 \text{ cubic feet}$$

$$P_t = \frac{P'}{8.07 \times 10^{-5}}$$

$$= \frac{2.88}{8.07 \times 10^{-5}}$$

$$= 35,670 \text{ ft-lb/sec}$$

To the value of the friction pressure drop Δp previously found should be added the pressure drop due to momentum increase as given by equation (36) and the end loss as given by equation (68), which is given in part III.

The selection chart (fig. 26) was used to construct the three-dimensional figures 27 and 28, which show the relations among the radiator variables at two altitudes for a pursuit airplane and for a heat dissipation rate of 474 Btu per second. Figures 27 and 28 should be of value in showing how a change in the value of any one of the variables affects the values of the others.

SELECTION OF INTERCOOLERS

Because the density of the atmosphere is a decreasing function of altitude, supercharging is used at altitude to compress the engine air in order that engine power can be maintained at or near its sea-level value. A supercharger, however, in compressing the air, also raises its temperature. The temperature of the air must then be reduced in order to increase its density and to prevent the detonation that results from high air temperatures. The reduction in temperature is accomplished by an intercooler, which is simply a heat exchanger in which both fluids are air.

Interrelations among Intercooler Variables

The selection of a heat exchanger presupposes that the design - that is, the internal geometry - has been chosen, that the rate of heat transfer which the heat exchanger must give is known, and that the altitude for which the heat exchanger is to be selected is known. The process of selection, then, consists in determining the external dimensions of the heat exchanger. For a given rate of heat transfer, these dimensions determine the pressure drops, the weight flow of cooling air, and the power cost of the exchanger.

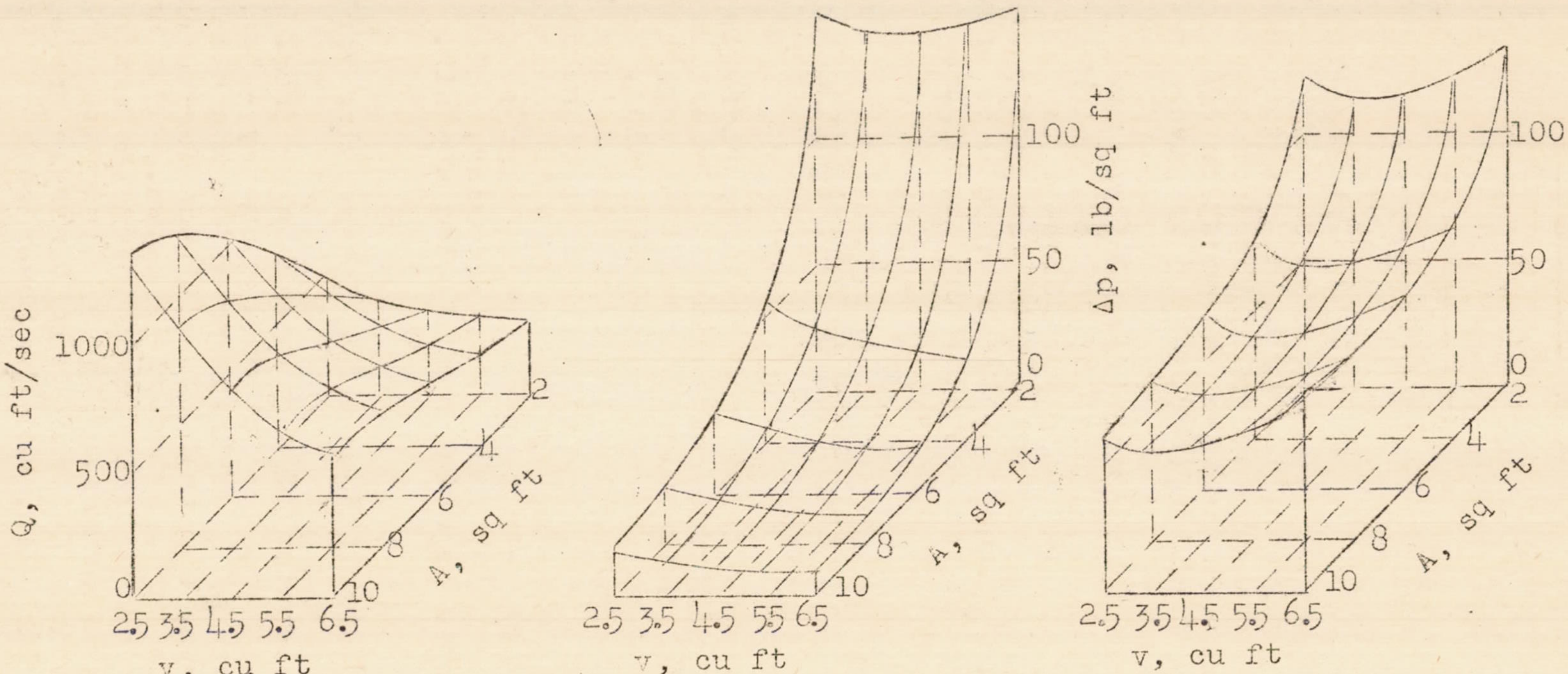


Figure 27. - Radiator variables for pursuit airplane. Altitude, 20,000 feet; heat dissipation, 474 Btu per second.

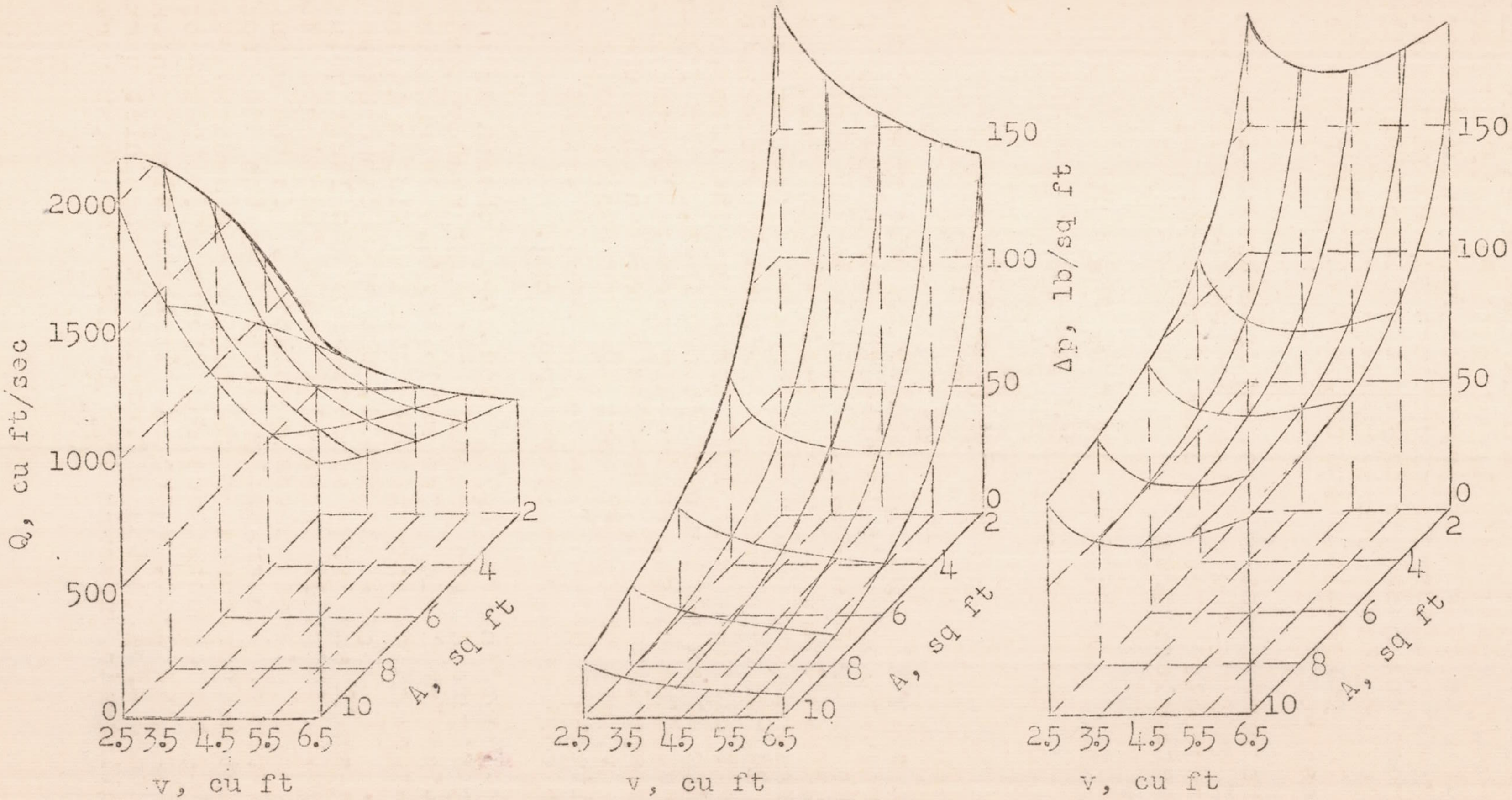


Figure 28. - Radiator variables for pursuit airplane. Altitude, 40,000 feet; heat dissipation, 474 Btu per second.

The selection of the dimensions must be so made that the five requirements previously listed will be met. In addition to being able to effect the required rate of heat transfer, the cooling-air pressure drop must not exceed that available, the dimensions must not exceed the dimensions of the available installation space, the engine-air pressure drop must not exceed the permissible amount, and the power cost should not be excessive.

The power cost P_t of an intercooler is taken as the cost of pumping the cooling air and the engine air through the intercooler plus the cost of transporting the weight of the intercooler; thus

$$P_t = Q_c \, \Delta p_c + Q_e \, \Delta p_e + P_W$$

where

$$P_W = \epsilon \frac{C_D}{C_L} \rho_R v \, V_0 \qquad (56)$$

In selecting an intercooler to meet some of the foregoing requirements, it is easy to obtain one that fails to meet the other requirements. In order to show how the different dimensions, the cooling-air weight flow, the cooling-air pressure drop, and the power cost affect one another, figure 29 has been prepared. Figure 29 was constructed for a pursuit airplane with 1675 normal engine horsepower operating at an altitude of 36,000 feet. The figure shows the three dimensions, engine-air passage length L_e, cooling-air passage length L_c, no-flow dimension L_n, the intercooler volume v, and the power cost P_t as functions of the ratio of cooling-air weight flow to engine-air weight flow W_c/W_e and the cooling-air pressure drop Δp_c.

The coordinates W_c/W_e and Δp_c chosen as a base for drawing figure 29 are probably the most nearly basic of the quantities involved. The ratio W_c/W_e determines the mean temperature difference between the two fluids in the intercooler. If W_c/W_e is small, the mean temperature difference is small and large surface area and large volume are required. (Fig. 12 shows $1/\zeta$ as a function of the ratio W_c/W_e. The dimensionless quantity ζ is the mean temperature difference in crossflow divided by the difference in the inlet temperatures of the engine air and the cooling air. In making the computations for figure 29, a value of $\xi = 0.736$ was used, where ξ is the ratio of engine-air temperature drop to the difference in the inlet temperatures.) The cooling-air pressure drop is also important because the

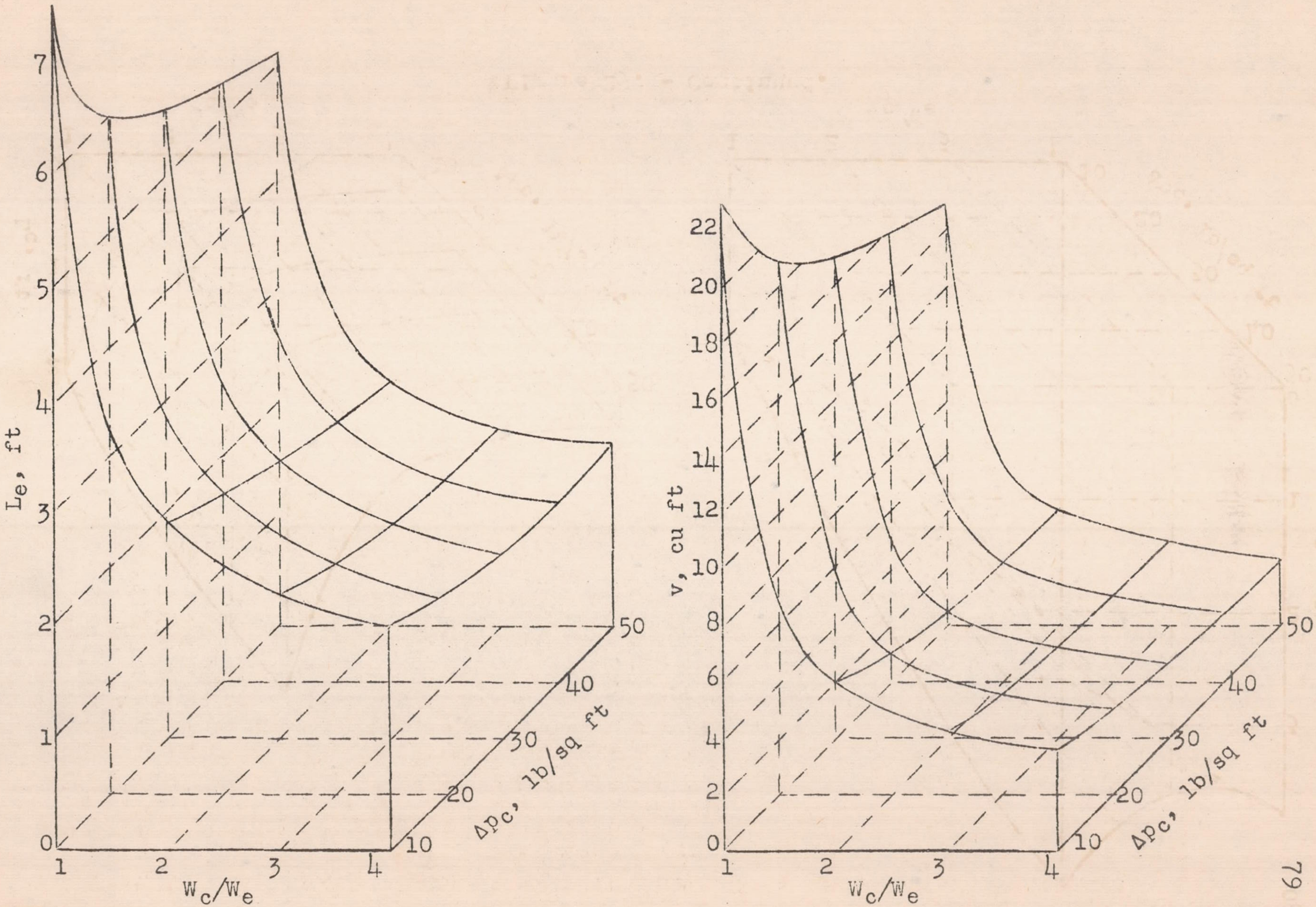
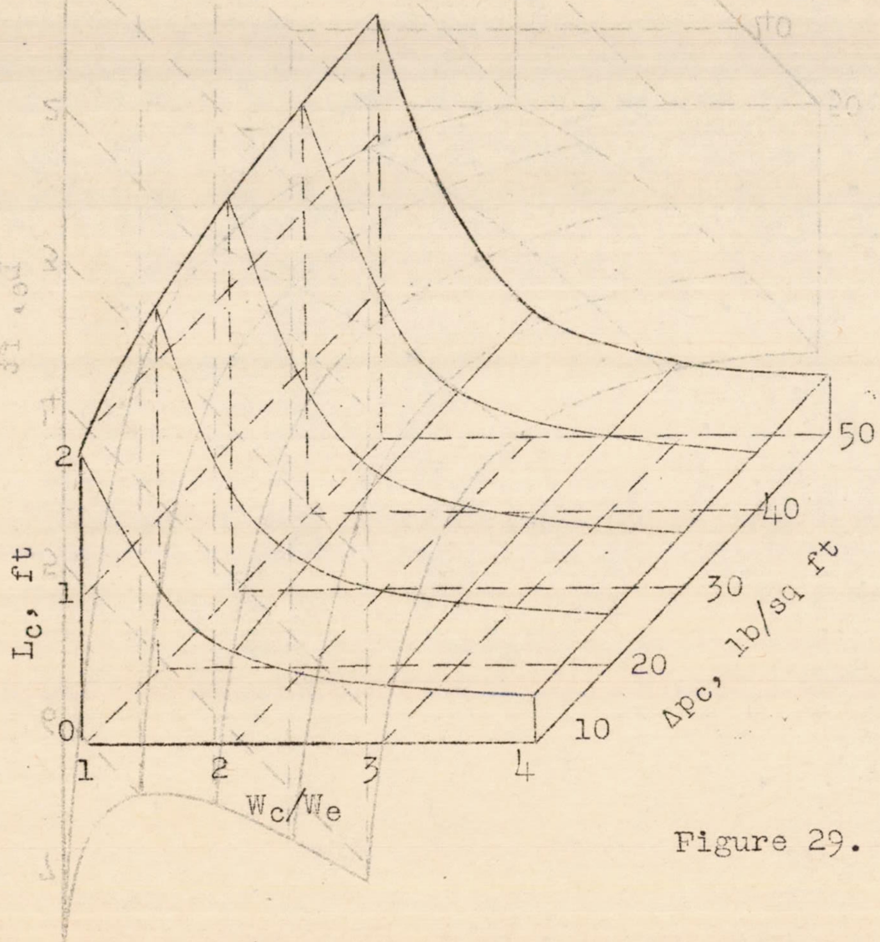


Figure 29. - Intercooler variables for pursuit airplane. Altitude, 36,000 feet.



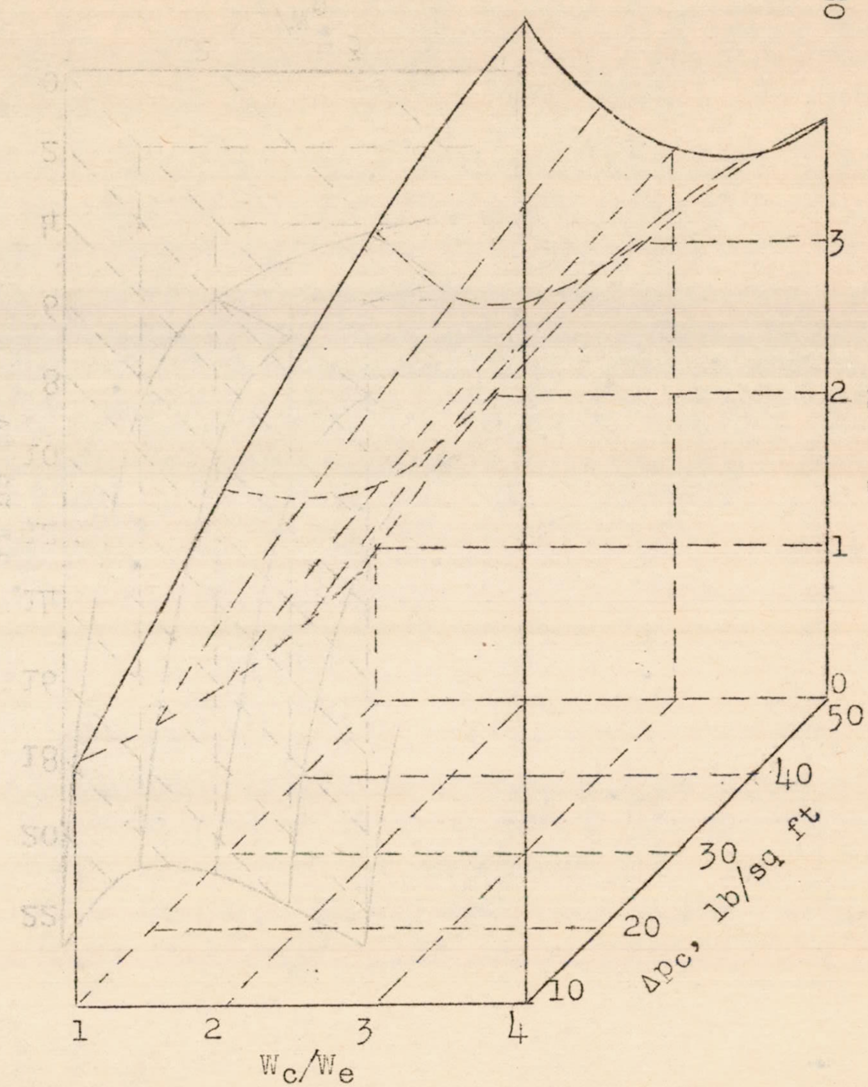


Figure 29. - Continued.

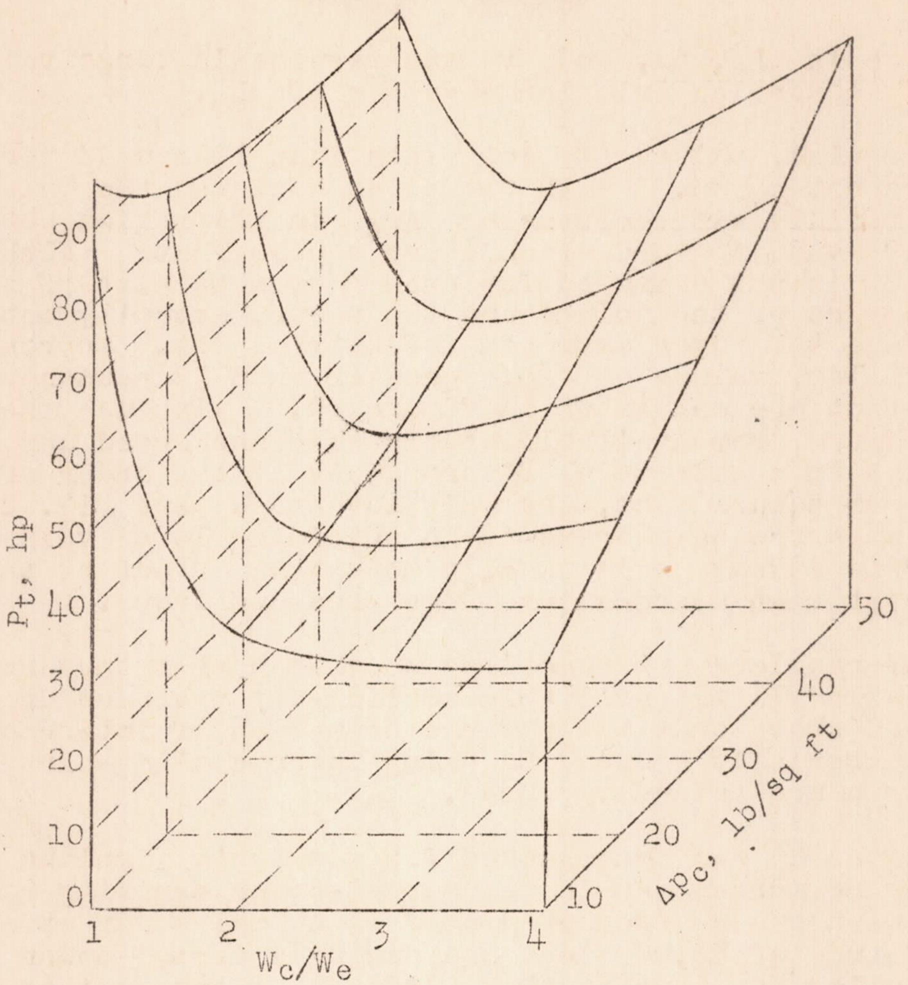


Figure 29. - Concluded.

upper limit is usually known (as some fraction of the free-stream dynamic pressure) and also because its value has a strong effect on the power cost of the intercooler.

Much information can be obtained from a study of figure 29. Only a limited part of the possible range of weight-flow ratio and cooling-air pressure drop is shown. In fact, most of the "bad" region is not shown. The lowest value of W_c/W_e used in the figure is unity. As previously stated, the value of W_c/W_e determines the mean temperature difference between the engine air and the cooling air. For values of W_c/W_e less than unity, the temperature difference is quite small (see fig. 12) and all the variables, except L_n, have exceedingly high values. Values of W_c/W_e greater than 4 also are not shown because the value of the mean temperature difference has leveled off with increasing W_c/W_e (fig. 12) and further increase in W_c/W_e produces negligible

decreases in L_e, L_c, and v and results in large values of P_t and in particularly large values of L_n.

Likewise, values of Δp_c less than 10 pounds per square foot are not shown. Both the power cost and the volume increase rapidly with decreasing Δp_c in the region between $\Delta p_c = 10$ and $\Delta p_c = 0$ pounds per square foot. Although figure 29 is not extended far enough even to hint of the rapid upturn of the volume and the power, actually both the volume and the power approach infinity as Δp_c approaches zero. Also, values of Δp_c greater than 50 pounds per square foot are not shown in figure 29. Although the cooling-air pressure drop available at the airplane speed and altitude used in constructing figure 29 is of the order of 100 pounds per square foot, the only advantage (see fig. 29) of using large pressure drops is in obtaining lower values of L_n. This advantage is usually more than offset by the large power cost that accompanies large pressure drops.

Intercoolers are sometimes so chosen that they require practically all the available cooling-air pressure drop. Figure 29 shows that the power consumption of intercoolers can be materially reduced by using lower cooling-air pressure drops than are frequently used.

Figure 29 was constructed for a pursuit airplane using 1675 engine horsepower in the normal-power cruising condition, with the engine supercharged to full normal power at a maximum altitude of 36,000 feet and having a normal-power speed of 460 miles per hour at that altitude. For engines of different power, L_n and v can be scaled in direct proportion to the engine power. The values of L_e, L_c, Δp_c, and W_c/W_e are independent of engine power.

Harrison aluminum louvered intercoolers with an engine-air pressure drop of 1 inch of mercury were used in calculating figure 29. For intercoolers of other design and for airplanes other than the one previously described, the figure, although not exactly applicable quantitatively, should nevertheless be quite useful in showing qualitatively the interdependence of the intercooler variables and how compromises among the variables can be effected.

Selection Charts

Enough experimental investigations have been made of different designs of intercooler to make it possible to write equations that correlate the heat transfer of all designs of

intercoolers yet constructed. It is of course possible to use these equations in selecting the external dimensions of intercoolers by calculating the rate of heat transfer, the pressure drops, and the power cost of a number of combinations of external dimensions and weight-flow rates. Such a procedure is, however, extremely tedious. The use of some form of selection chart is much more desirable. Two general types of selection chart are available for selecting intercoolers; namely, performance charts and generalized selection charts.

Performance charts are made up, usually by the manufacturer, for intercoolers of a given internal geometry. It is customary to construct, from test data, a set of charts for different combinations of dimensions and pressure drops. When a set of these charts is available, the selection of an intercooler that satisfies most of the requirements is relatively easy. This nongeneralized type of chart, however, is not suitable for showing the power cost of the intercoolers and does not easily show the effects on all the other variables when any one variable is systematically varied.

The second type of chart is the generalized chart. A single chart can be constructed that is applicable for all intercoolers of a given general design; that is, one generalized chart can be used for all tubular intercoolers of all tube sizes and tube spacings. Another single chart can be used for all intercoolers of the Harrison type, no matter what the size of the passages. Such charts are particularly valuable in that they show how the other variables are affected when one of them is systematically varied. Because the power cost is included as one of the variables shown on the generalized chart, the selection of the most satisfactory intercooler for a given installation generally requires only one or two trials with the generalized chart but may mean a large number of trials with performance charts.

Generalized selection charts for tubular intercoolers and for Harrison intercoolers are given in reference 17. These charts are highly generalized in that a single chart can be used for selecting all intercoolers of a given general design, for all airplanes, and for all altitudes. Because of the high degree of generalization, the auxiliary calculations that must be made in the selection of an intercooler with these charts are tedious. For intercoolers of a given internal geometry, however, many of these calculations can be made once, the results incorporated into a selection form, and the form used to simplify the selection process. Selection forms for an Airesearch tubular intercooler and for

Harrison copper intercoolers are presented in reference 17 and for Harrison aluminum intercoolers in reference 18.

The selection of an intercooler always represents a compromise among dimensions, cooling-air pressure drop, and power cost. Generalized charts make it possible to select, usually on the first trial, the intercooler that represents the most satisfactory compromise for any given set of conditions. In fact, without the aid of the generalized charts, it is extremely difficult to select, for any given set of conditions, the best intercooler. On the other hand, for one who is not familiar with the generalized charts, the selection of an intercooler by a performance chart is undoubtedly easier than by the generalized charts. The best recommendation therefore seems to be that the performance charts be used for choosing intercoolers and that the selection be guided by figure 29.

Preliminary Calculations

A number of preliminary calculations have to be made before a selection chart is used. In the present section, these calculations are illustrated for a typical case. An airplane having an engine with 1200 normal brake horsepower is assumed. The engine is supercharged by a turbosupercharger. The critical altitude, that is, the maximum altitude at which full normal engine power is developed, is 20,000 feet. The level-flight normal-power speed at that altitude is 400 miles per hour.

Intercoolers are usually selected for the altitude at which the cooling problem is the most difficult. For gear-driven superchargers, this altitude is usually the one at which the gears are shifted into the highest ratio. The quantity ξ, which is the drop in engine-air temperature required of an intercooler divided by the difference in the engine-air and the cooling-air inlet temperatures, is plotted against altitude for a gear-driven supercharger in figure 30.

For turbosuperchargers, the most severe demands are made on the intercooler at the critical altitude. Figure 31 illustrates the variation of ξ with altitude for a turbosupercharger installation.

If the supercharger compressed the air isentropically, the ratio of the air temperatures at the outlet and at the inlet of the compressor T_o/T_i would be given by the

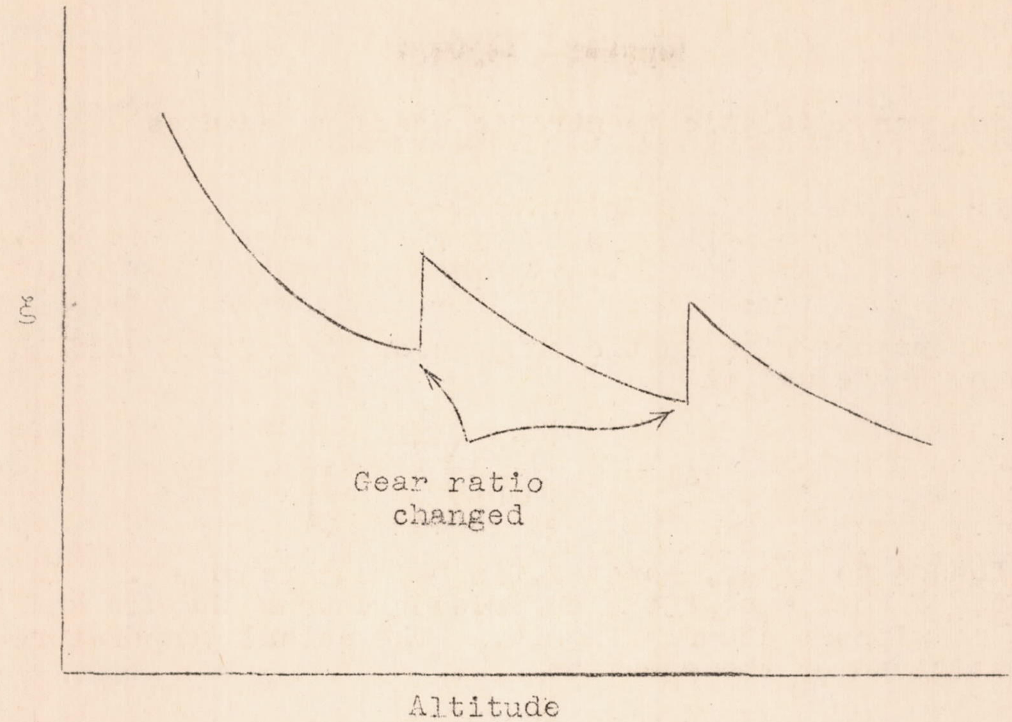


Figure 30. - Effect of altitude on ratio of drop in temperature of hot fluid to inlet temperature difference ξ for intercooler. Gear-driven supercharger.

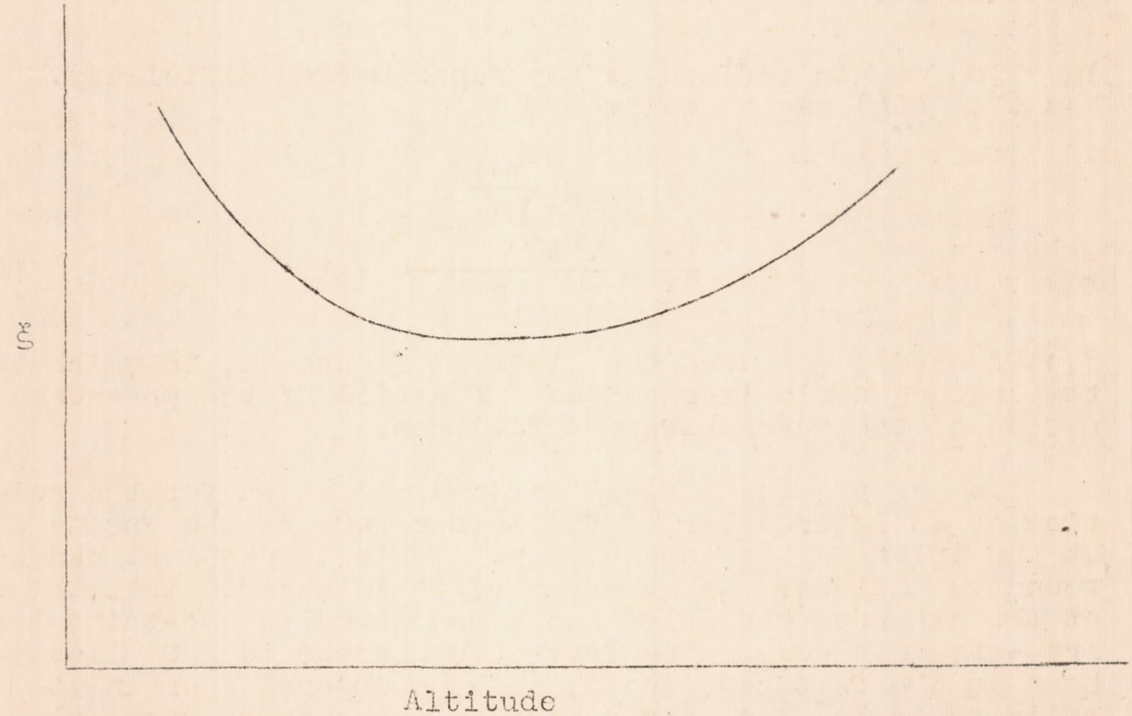


Figure 31. - Effect of altitude on ratio of drop in temperature of hot fluid to inlet temperature difference ξ for intercooler. Turbosupercharger.

equation for adiabatic isentropic pressure changes

$$\frac{T_o}{T_i} = \left(\frac{p_o}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \tag{50}$$

The temperature rise in the compressor therefore would be given by the equation

$$\Delta T = T_i\left[\left(\frac{p_o}{p_i}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

Losses do occur, however, in the supercharger. In effect, to some extent the compressor churns the air and heats it without compressing it. The actual temperature rise is given by the equation

$$\Delta T = \frac{T_i\left[\left(\frac{p_o}{p_i}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta} \tag{66}$$

in which η is defined as the supercharger efficiency. Equation (66) can be written

$$\frac{T_o}{T_i} = \frac{\left(\frac{p_o}{p_i}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta} + 1 \tag{67}$$

Equation (67) was used for plotting figure 32, in which the temperature ratio is shown as a function of the pressure ratio and the supercharger efficiency.

One of the quantities that must be known for the selection of an intercooler is the temperature of the engine air at the inlet to the intercooler. This temperature can be found as follows: As an example, it is assumed that the air at the supercharger inlet has been slowed to one-third its free-stream speed. The free-stream speed is 400 miles per hour or 588 feet per second, and the supercharger inlet velocity is 196 feet per second. The supercharger inlet

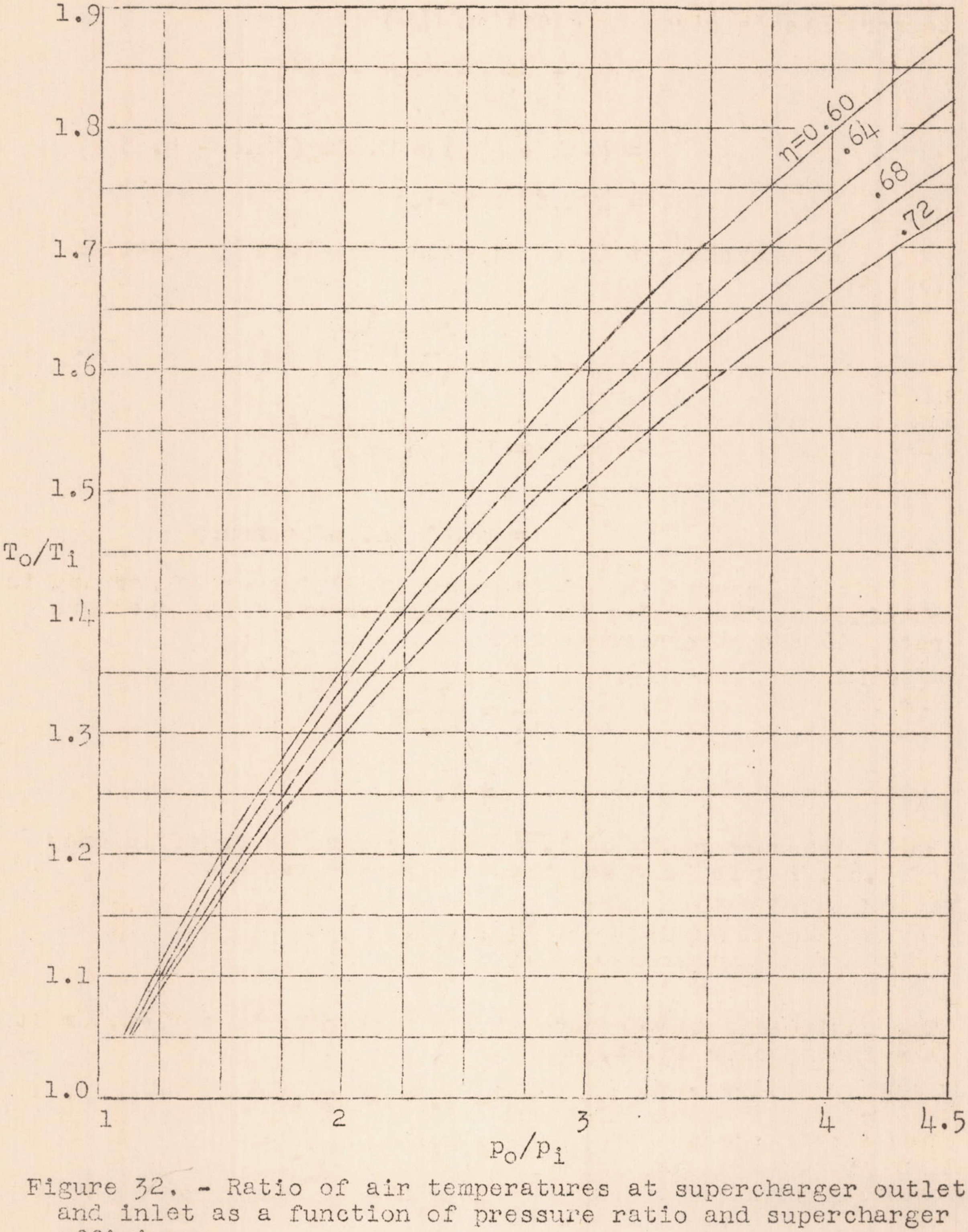


Figure 32. - Ratio of air temperatures at supercharger outlet and inlet as a function of pressure ratio and supercharger efficiency.

temperature is given by equation (49) as

$$T_i = T_0 + \frac{0.832}{10^4}(V_0{}^2 - V_i{}^2)$$

$$= (-12 + 459) + 0.832\ (34.6 - 3.8)$$

$$= 472.6^\circ \text{ F abs.}$$

The supercharger inlet pressure is given by equation (50) as

$$p_i = p_0 \left(\frac{T_i}{T_0}\right)^{\frac{\gamma}{\gamma-1}}$$

$$= 13.75 \left(\frac{472}{447}\right)^{3.5}$$

$$= 16.63 \text{ in. of mercury}$$

It is assumed that a pressure of 31 inches of mercury is obtained at the outlet of the supercharger. The pressure ratio in the supercharger then is

$$\frac{p_o}{p_i} = \frac{31}{16.63}$$

$$= 1.86$$

For a pressure ratio of 1.86 and a supercharger efficiency of 0.60, figure 32 gives the temperature ratio as

$$\frac{T_o}{T_i} = 1.31$$

The engine-air temperature at the supercharger outlet, or at the intercooler inlet, then is

$$T_o = 1.31 \times 472.6$$

$$= 619^\circ \text{ F abs.}$$

$$= 160^\circ \text{ F}$$

For illustration, the engine-air temperature required at the intercooler outlet is assumed to be 80° F. The engine-air temperature drop required in the intercooler therefore is

$$\Delta T = 160 - 80$$

$$= 80^\circ \text{ F}$$

In order to find ξ, the inlet temperature of the cooling air must be found. It can be estimated by estimating the adiabatic temperature rise that the cooling air undergoes between the free stream and the intercooler inlet. The free-stream speed is 400 miles per hour or 588 feet per second. The estimated speed of the air in the intercooler is 75 miles per hour or 110 feet per second. The adiabatic temperature rise, as given by equation (49), therefore is

$$\Delta T = \frac{0.832}{10^4}\left[(400)^2 - (110)^2\right]$$

$$= 28^\circ \text{ F}$$

The cooling-air inlet temperature then is

$$T_c = -12 + 28$$

$$= 16^\circ \text{ F}$$

The difference in inlet temperatures is

$$160 - 16 = 144^\circ \text{ F}$$

The engine-air temperature drop divided by the inlet temperature difference is

$$\xi = \frac{80}{144}$$

$$= 0.56$$

The rate of engine-air flow can be calculated by multiplying the engine brake horsepower by the rate of air consumption per engine horsepower. The rate of air consumption is furnished by the engine manufacturer and is of the order of 7.5 pounds per hour per horsepower. For the present example, then, the engine-air flow is

$$W_e = \frac{1200 \times 7.5}{3600}$$

$$= 2.5 \text{ lb/sec}$$

The rate of heat transfer required is given by equation (4) as

$$H = W_e \, c_p \, \Delta T \tag{4}$$

$$= 2.5 \times 0.24 \times 80$$

$$= 48.0 \text{ Btu/sec}$$

After these preliminary calculations are made, an intercooler may be selected. Any one of a number of methods or charts can be used - either the performance charts published by the various intercooler manufacturers such as those in reference 19, or the charts published in reference 20, or the generalized charts of reference 17. A detail description of the use of these charts is not desirable in the present report. The mechanics of each method is clearly explained in the reference report. In the present report, it is desired only to suggest that the picture presented by figure 29 be kept in mind in order that a good selection may be made.

SELECTION OF OIL COOLERS

Oil coolers are used to dissipate a part of the heat developed in engines and to keep the oil at a safe temperature. The heat rejection to the oil is usually specified by the engine manufacturer. It is customarily taken as about 6 or 7 percent of the brake horsepower for air-cooled engines and about 9 or 10 percent of the brake horsepower for liquid-cooled engines. The heat rejection to the oil may vary not only from engine to engine but also with the age of the engine, inasmuch as it depends somewhat on the clearances in the engine.

As was pointed out in the discussion of oil coolers in the section entitled "Applications to Design," in part I, the Reynolds number of the flow of oil in coolers is so low that the flow is laminar rather than turbulent. When the temperature of the cooling air is low, the temperature of the oil may become so low that the oil congeals. When this condition occurs, the thermal resistance on the oil side becomes so great that the oil which may be forced through the cooler is not properly cooled. An example of the effect of congealing in a typical oil cooler is shown in figure 33. As the inlet air temperature is decreased, the rate of heat dissipation increases until a certain air temperature is

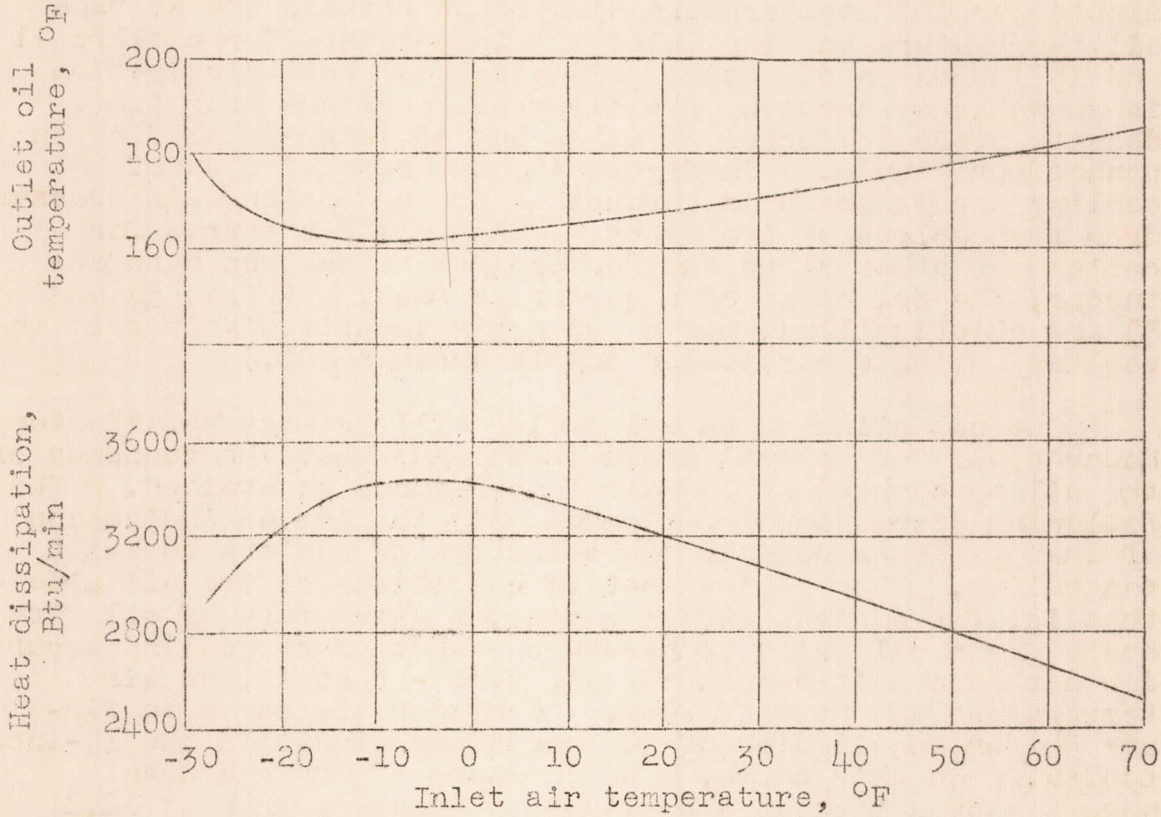


Figure 33. - Effect of congealing in oil cooler on rate of heat dissipation and on oil outlet temperature.

reached - in this instance, about -5° F. Below this temperature, heat dissipation decreases and oil outlet temperature increases because the oil is congealing.

The satisfactory correlation of heat transfer in highly viscous fluids, such as oil, in laminar flow in smooth straight tubes is not yet possible. Correlation for oil coolers is further complicated by congealing, by aeration of the oil, and by the different heat-transfer characteristics of the baffle systems used by different manufacturers. For these reasons, generalized selection charts, such as those discussed previously for coolant radiators and air intercoolers, are not yet available for oil coolers. Performance charts have been constructed, however, by the various manufacturers and can be satisfactorily used for the selection of oil coolers.

The performance charts are usually set up in a form somewhat similar to that used in figures 34 and 35. In figure 34, the rate of heat dissipation is shown in Btu per minute per 100° temperature difference between the average oil temperature and the inlet air temperature for coolers 11 and 15 inches in diameter. This rate of heat dissipation is shown as a function of cooling-air pressure drop times density ratio in inches of water and of rate of oil flow in pounds per minute. In figure 35, the rate of flow of cooling air is shown as a function of the cooling-air pressure drop for coolers of two sizes. With similar charts for coolers of other sizes and for coolers of various manufacturers, the selection of a cooler is easy. (Figs. 34 and 35 are chiefly illustrative and apply quantitatively only to coolers having a particular baffle arrangement.)

The selection of an oil cooler will be most satisfactory, however, if the selection can be so made that the tendency of the oil to congeal at low air temperatures is avoided. The following discussion is intended only to give an indication of what might be done in the selection of coolers to avoid congealing. For a given set of conditions on the oil side - that is, for a given baffle system, a given rate of oil flow, and a given oil inlet temperature - whether congealing occurs depends on conditions on the air side - that is, on air temperature and rate of flow. Consider the curve in figure 34 for an oil flow of 90 pounds per minute in the 15-inch cooler. On this curve, a temperature of 60° has been marked with a tick at a cooling-air pressure drop of 12.2 inches of water to indicate that, if the inlet air temperature is 60° F, congealing is likely to occur if an air pressure drop greater than 12.2 inches of water is used. In other words, the part of the curve to the left of the tick marked 60° can be used without danger of congealing, if the inlet air temperature is 60° F or above. The part of the curve to the right of the 60° tick is unsafe for use if the inlet air temperature is 60° F or less. Correspondingly, the tick marked 40° at a pressure drop of 3.3 inches of water indicates that, on that part of the curve to the right of 3.3 inches of water, congealing will occur if the inlet air temperature is 40° F or less. Similarly, temperatures are marked with ticks on the curves for other oil flows in the 15-inch cooler.

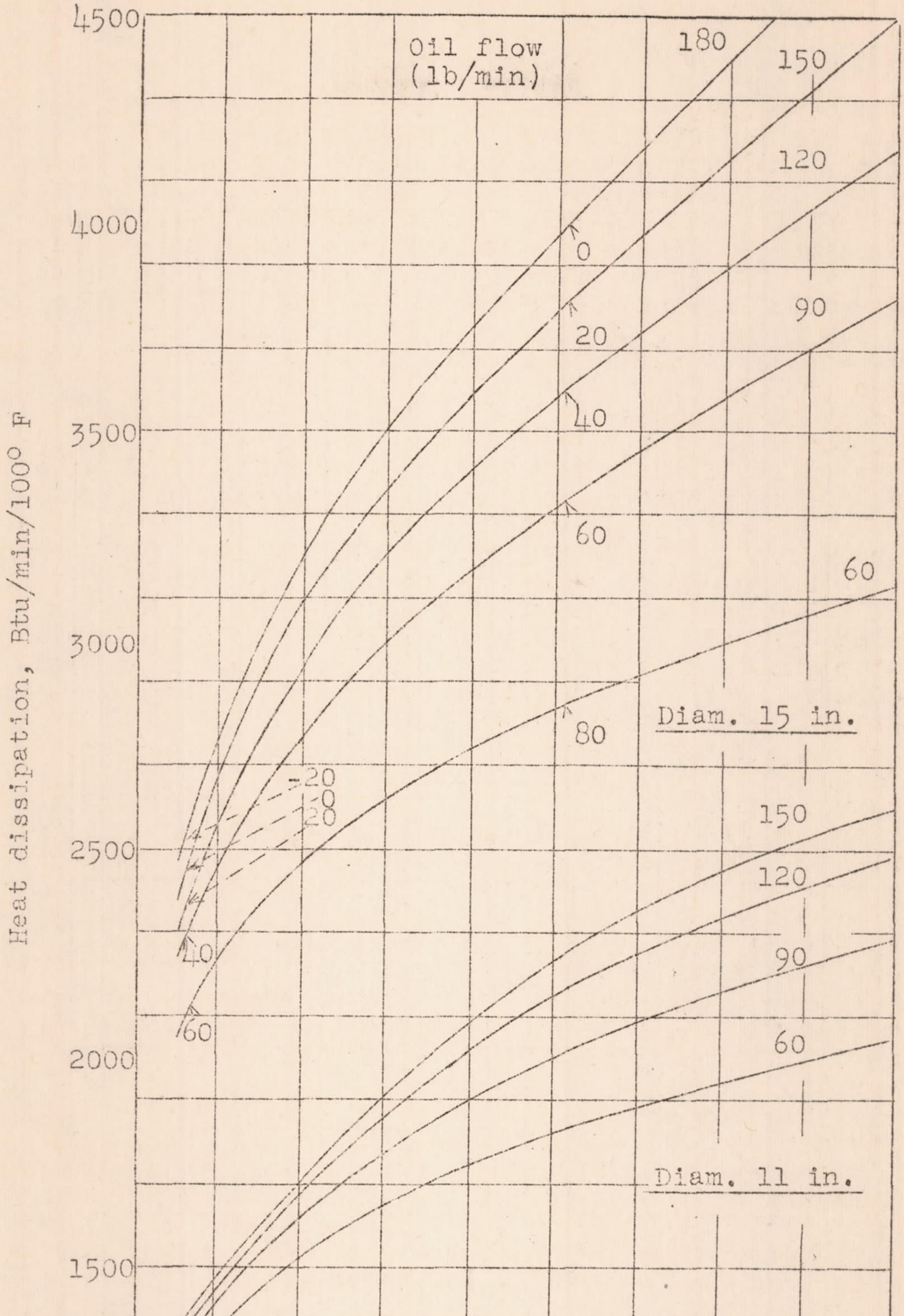


Figure 34. - Typical oil-cooler performance chart - heat dissipation as a function of cooling-air pressure drop and oil flow rate.

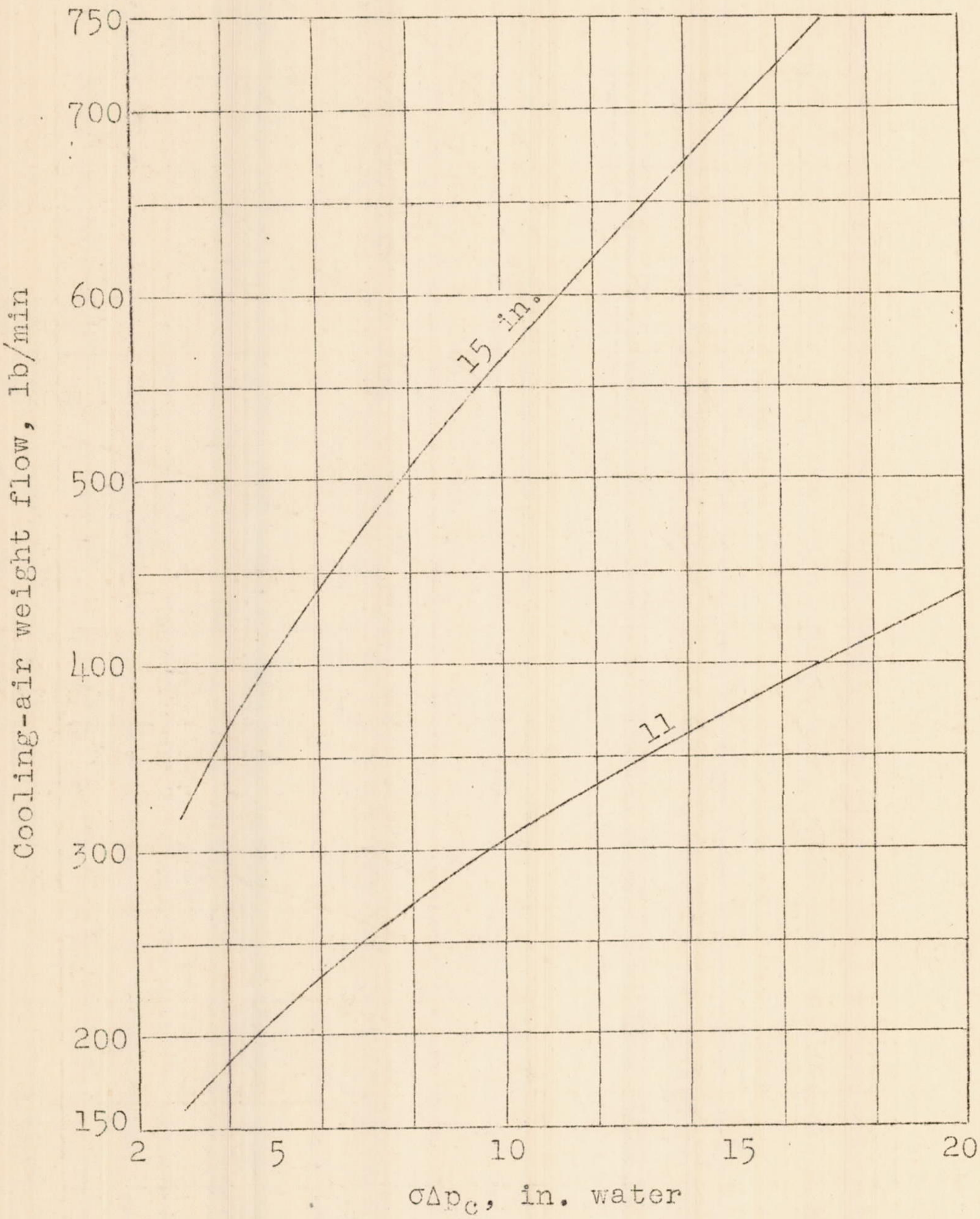


Figure 35. - Cooling-air weight flow as a function of cooling-air pressure drop for cooler of figure 34.

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SELECTION OF ENGINE FINS

If an engine is not adequately cooled, improvements in the cooling will make possible a number of improvements in the performance of the engine. Better cooling makes possible a decrease in fuel consumption, an increase in the power output that can be used, a reduction in detonation, and an increase in the altitude at which sufficient pressure drop for cooling is available. As examples of the improvements that can be made in the cooling of air-cooled engines by relatively simple changes in the finning, the following illustrations are presented. These illustrations show how the maximum altitude at which full power can be taken from an engine, as determined by the cooling of the engine, can be increased by increasing the fin surface area by the use of wider and thinner fins.

It has previously been shown that the rate of heat transfer from a finned air-cooled cylinder is a function of the fin width, thickness, and spacing, of the thermal conductivity of the fin metal, and of the ρV of the cooling air in the fins. At high altitudes, the cooling of an engine usually becomes difficult because the low air density makes a high air velocity necessary. The high velocity results in a large pressure drop. The pressure drop required for cooling may be greater than that available. In such a case, the velocity required to effect the necessary rate of heat transfer, and accordingly the pressure drop required, can be reduced by increasing the fin surface area. As illustrations, consider figure 36, which is taken from figure 16 of reference 12. In figure 36, the ratio of the engine cooling-air pressure drop required to the pressure drop available, in the high-speed-flight condition of the airplane considered in reference 12, is plotted as a function of altitude for four different fin arrangements. It is assumed that the engine develops full normal horsepower at all the altitudes considered. The engine cannot be cooled when the ratio of pressure drop required to pressure drop available exceeds unity. For aluminum fins 0.75 inch wide and 0.06 inch thick - dimensions representative of the finning on old or low-powered engines - the limiting altitude is 25,000 feet. With aluminum fins 1 inch wide and 0.06 inch thick, the engine can be cooled to 42,000 feet. With aluminum fins 1.5 inches wide and 0.052 inch thick, the limiting altitude is raised to 55,000 feet. With copper fins 1.5 inches wide and 0.035 inch thick, the engine can be cooled at altitudes up to 59,000 feet.

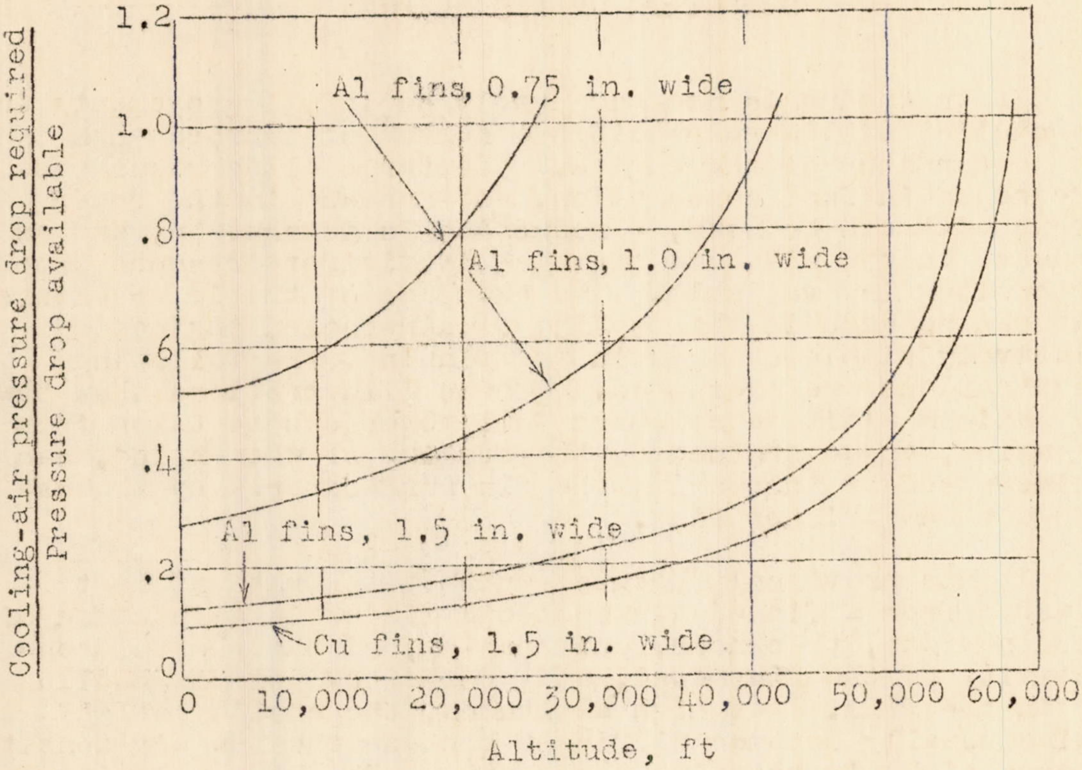

Figure 36. - Effect of fin dimensions and altitude on cooling of radial engine. (From fig. 16 of reference 12.)

The foregoing examples of improving engine cooling, which are confined to increasing the fin width and decreasing the fin thickness, are not exhaustive of the possibilities of improving cooling by changes in fin dimensions. The pressure drop required for cooling can also be materially reduced by the use of smaller fin spacing, for example. As further illustrations of possible improvements, consider the sources of pressure loss across an engine as shown in figure 37. This figure, which is taken from figure 15 of reference 12, gives a breakdown of the pressure drop shown in figure 36 for the engine with the 1-inch fins. As is shown, the pressure drop is made up of three components: (1) the drop caused by friction in the fins, (2) the drop that accompanies the increase in the momentum of the air, which is caused by the change in the temperature and the pressure of the air, and (3) the loss that is caused by the abrupt expansion at the exit from the baffles. Only the first of these components, that due to friction, should be considered a necessary and useful pressure drop. The other two components should be considered not only not useful but also not necessary. An arrangement in which pressure loss is caused only by friction

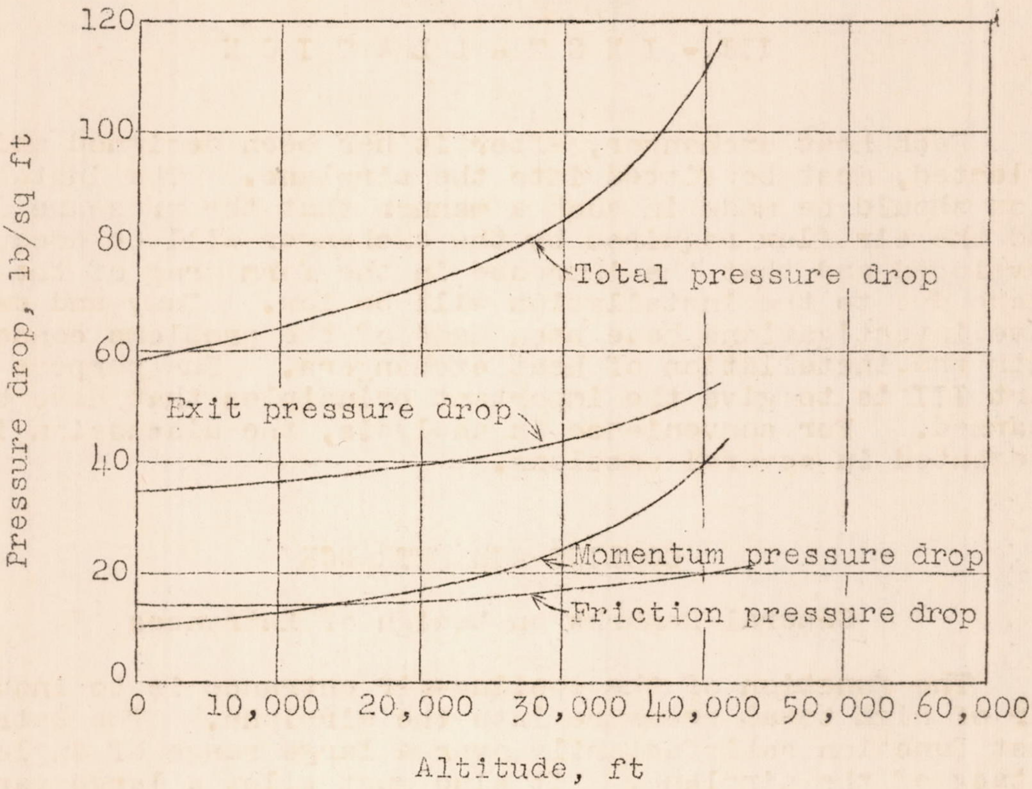

Figure 37. - Breakdown of cooling-air pressure drop across radial engine. Friction pressure drop, momentum pressure drop, exit pressure drop, and total pressure drop for 1-inch fins as functions of altitude. (From fig. 15 of reference 12.)

should be kept in mind as the ideal arrangement. As is pointed out in reference 21, if the air passages can be designed to have the proper expansion from entrance to exit, with the result that the dynamic pressure of the air remains constant in spite of a change in its density, then the momentum of the air remains constant and the pressure loss due to increase in momentum is eliminated.

The large loss that is due to the abrupt expansion at the exit from the baffles, furthermore, is proportional to the dynamic pressure at the exit. By the use of wider fins, the exit dynamic pressure is reduced and the expansion loss accordingly is reduced. Furthermore, by an improvement in the exit, such as the provision of a gradually expanding duct following the baffle, the loss due to the expansion can be reduced.

III - INSTALLATION

Each heat exchanger, after it has been designed and selected, must be fitted into the airplane. The installation should be made in such a manner that the pressure drop and the air flow required by the exchanger will be adequately developed and that the increase in the form drag of the airplane due to the installation will be low. Many and extensive investigations have been made of the problems connected with the installation of heat exchangers. The purpose of part III is to give the important principles that have been learned. For convenience in analysis, the discussion is presented in several sections.

COOLING-AIR ENTRANCE

General Remarks on Design of Entrances

The function of the cooling-air entrance is to induct air of high total pressure into the airplane. The entrance must function satisfactorily over a large range of angle of attack of the airplane. It also must allow a large variation in the volume rate of flow as the altitude and the engine power change. The leading edge of the entrance therefore must have the same characteristics as the leading edge of a wing. If a wing were designed to fly always at one angle of attack, the lowest drag would be obtained with a relatively sharp leading edge properly alined for that angle of attack. A practicable wing must be capable of functioning through a large variation of angle of attack, however, and must therefore have a well-rounded leading edge. For the same reasons, the leading edge of an entrance must also be well rounded.

Figure 38 shows the variation with altitude of quantity of cooling air Q for an air-cooled engine having a rated power of 1675 horsepower (reference 12). It can be seen that the velocity of the air in the entrance varies in the ratio of about $2\frac{1}{2}$:1 between an altitude of 40,000 feet and sea level.

The power developed by the engine obviously has a marked effect upon the cooling requirements. Figure 39 illustrates the variation of the ratio of the quantity of cooling air required to the quantity required for rated engine power output as a function of the engine power. Under operating conditions in flight, engines operate all the way from about 33 percent to more than 100 percent of normal power.

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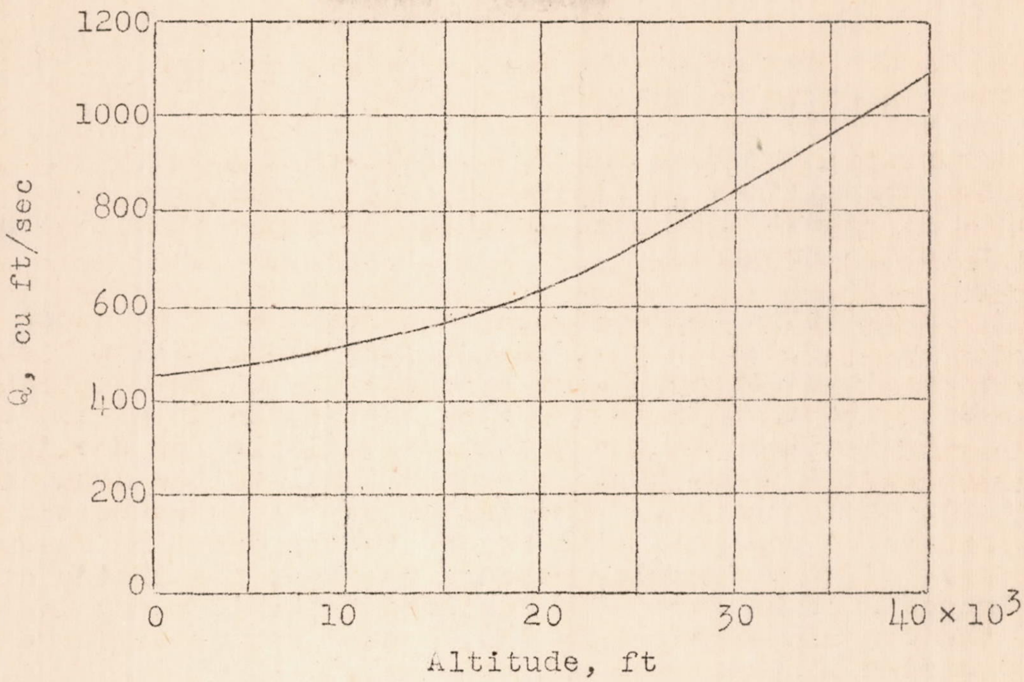


Figure 38. - Quantity of cooling air required by 1675-horsepower engine as a function of altitude. (Calculated data of reference 12.)

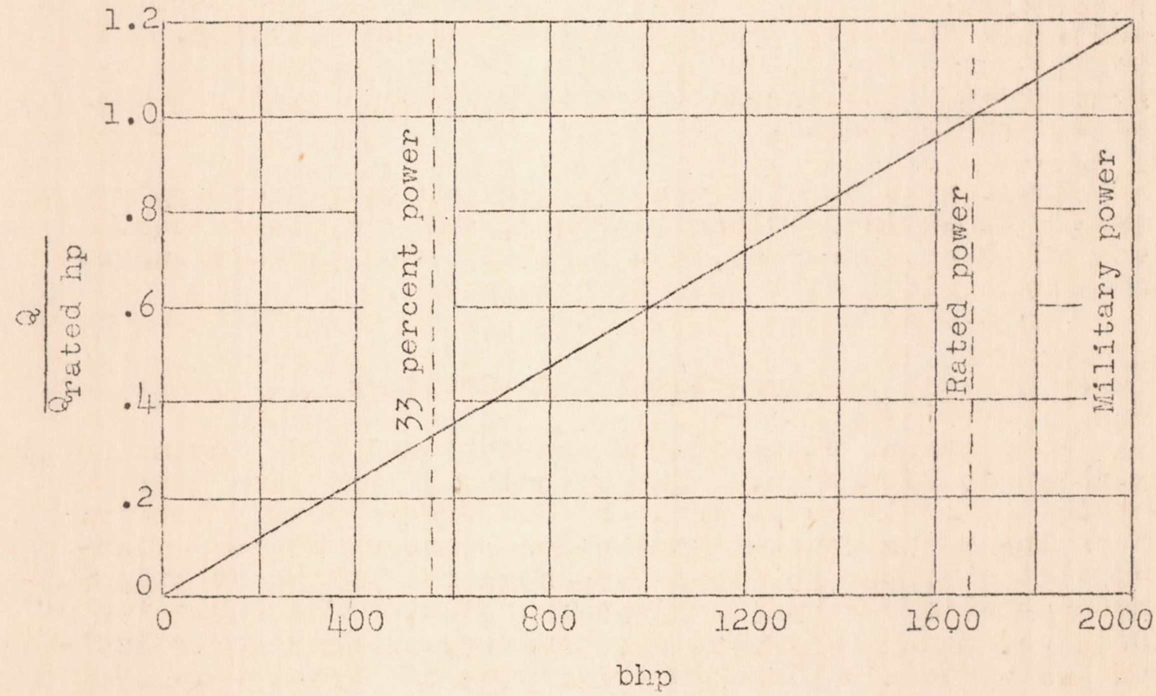


Figure 39. - Effect of engine power on quantity of cooling air required.

In the design of the leading edge, however, not the actual inlet velocity of the cooling air but the ratio of inlet velocity to airplane velocity is the important factor. The variation of this factor is illustrated in the variation in the streamlines in figure 40, which shows graphically the range of conditions under which an entrance should be able to function properly.

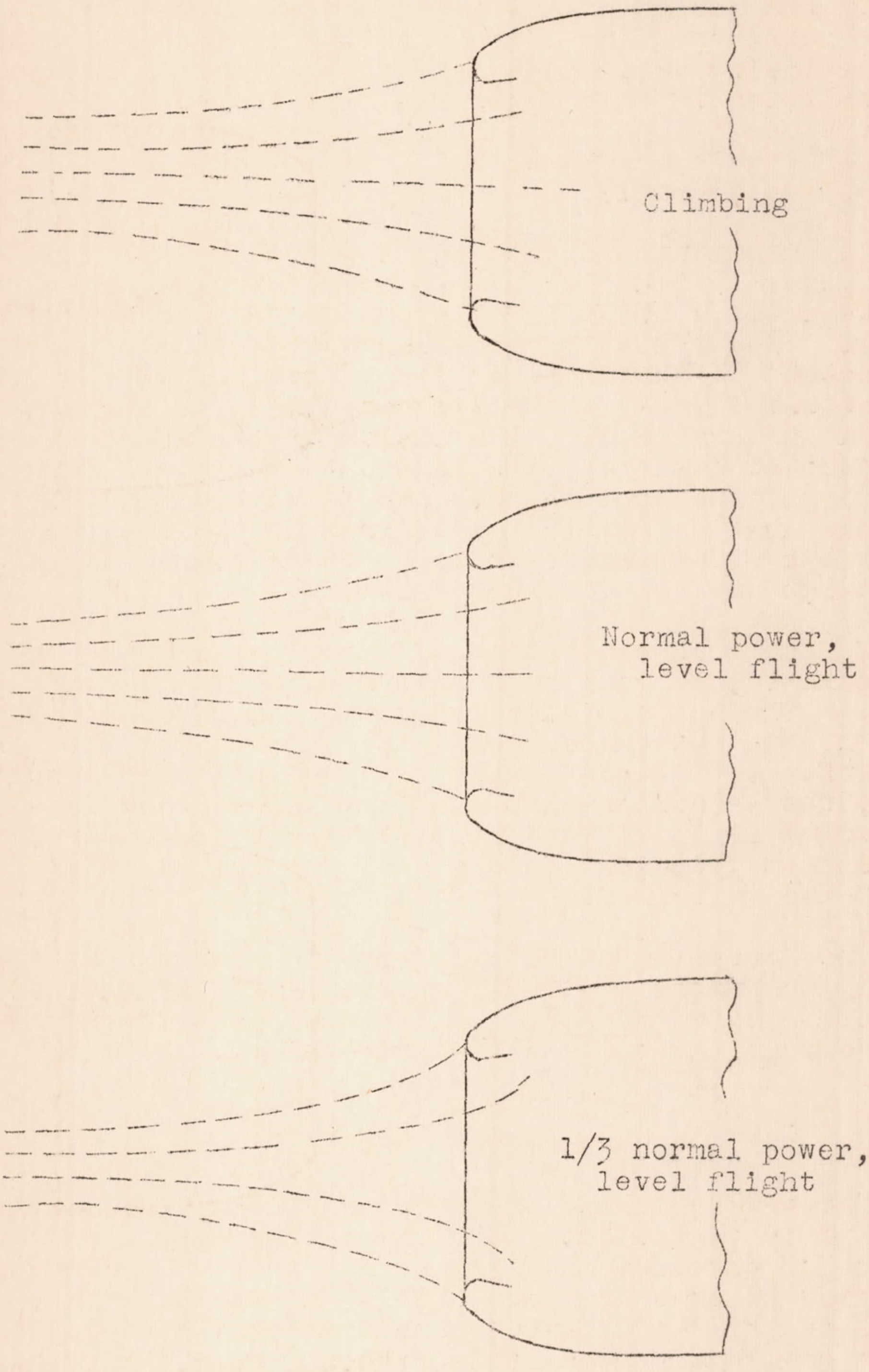
In addition to developing the required flow under a wide range of conditions, an entrance should allow the flow to take place without large external or internal entrance losses. Separation of the flow on the inside of the entrance will cause a loss in the pressure available for cooling and an increase in drag that is proportional to the dynamic pressure in the entrance. The inside of the entrance should therefore be designed with regard to the dynamic pressure there. If the dynamic pressure is high, the inside of the entrance must be carefully designed to avoid excessive losses. If the dynamic pressure is low, losses cannot be large and the design is therefore not critical.

If there is separation of the flow over the outside of the entrance, large losses that result in high drag will occur. The shape of the outside of the entrance behind the leading edge determines the local velocity of the air over the surface. If the radius of curvature of the surface is large, the velocity approaches free-stream velocity. If the radius is small, the velocity is large compared with free-stream velocity and the drag may then be high because a shock wave is formed.

The foregoing statements regarding entrances apply equally to engine cowlings, scoops, and wing entrances. Each of these entrances, however, has peculiar advantages and shortcomings that must be recognized.

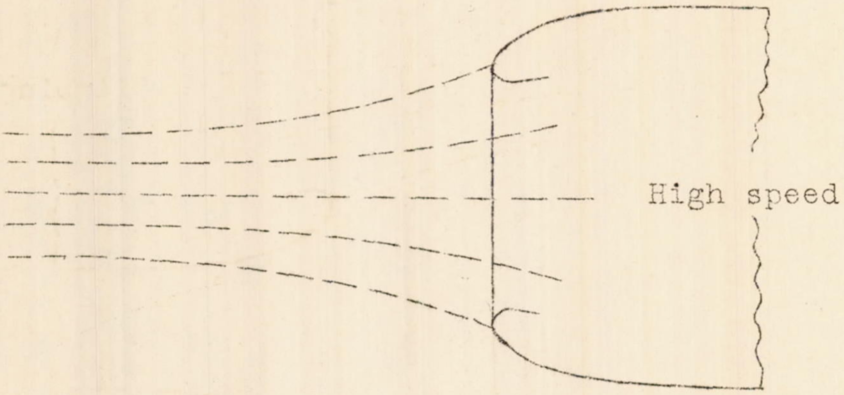
Conventional NACA Cowlings

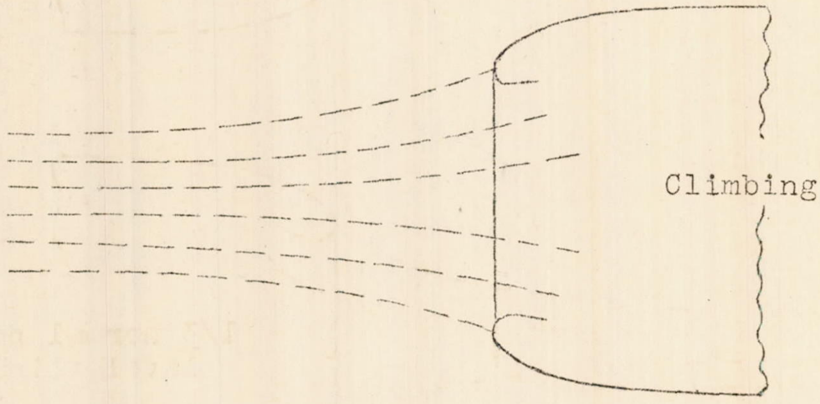
The general shape of the conventional NACA engine cowling is shown in figure 40. The velocity of the flow over the outside of cowlings is greater than free-stream velocity. According to pressure-distribution measurements and wind-tunnel tests, the critical Mach number of the conventional engine cowlings generally is rather low. The formation of shock waves may be prevented, however, as previously indicated, by modifications in the radius of curvature. Up to the present time, no airplane has experienced large drag increases due to the formation of shock waves on a conventional



(a) At sea level.

Figure 40. - Variation in cross section of cooling air in free stream for various operating conditions.





(b) 40,000 feet.

Figure 40. - Concluded.

cowling. Shock waves form on some of the other important parts of an airplane at lower airplane speeds than on the conventional NACA cowling. The most critical parts, in decreasing order of importance, are: propeller, wing-fuselage juncture, cowling, and wing. High drag from conventional cowlings has resulted, as is discussed later, principally from a poor design of the afterbody of a cowling or from a poor alinement of the afterbody and the wing.

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Scoops

The greatest disadvantage of scoops is that they add to the frontal area of the airplane. Air taken in by a scoop generally can be more economically inducted through an opening in the front of a nacelle or in the leading edge of a wing. Furthermore, scoops are usually located where a boundary layer already exists on the airplane. An entrance that admits air of both high and low energy tends to allow the air to flow out of the scoop at the corners where the scoop meets the surface of the airplane. If the air is slowed down considerably before it enters the scoop, the rapid pressure rise may cause a separation of the boundary layer just in front of the entrance. Such a separation is accompanied by a large entry loss, by nonuniform pressure distribution in the entry, and by spillage or return flow from the corners of the scoop. One remedy is to raise the scoop off the surface in such a way that the low-energy air is diverted around the scoop. Another remedy is to decrease the entrance area of the scoop in order that the air will enter at five-tenths to seven-tenths airplane speed. The first method somewhat increases the frontal area of the scoop. The second method results in an entrance that is critical to the changes in flow associated with changes in engine power and in altitude and also necessitates care in the design of the inside of the entrance, on account of the relatively high dynamic pressure that must be handled.

Wing Entrances

Wing entrances meet some of the conditions for desirable entrances. They are located at a forward stagnation point on a well-streamlined body. The wing behind the entrance provides a desirable housing for the heat exchanger. The chief consideration in the design of wing entrances is so to arrange the entrance that the flow over the wing, and consequently the lift, is not appreciably affected. The solution of this problem requires care and experience. For structural

reasons, a wing is designed to be as thick as possible and therefore a large expansion of the flow over the after part of the wing is required. At high angles of attack, the expansion becomes critical. Any interruption of the flow near the leading edge such as is caused by an entrance, guns, bumps, and so forth, tends to produce an instability in the flow over the wing that causes separation in the region of expansion. Separation results in increased drag, reduced lift, and reduced maximum lift coefficient.

Another factor that merits consideration when air is taken into an airplane at the leading edge of a wing is the change in the orientation of the entry with respect to the local air flow when the angle of attack of the wing is changed. Because of the increase in the upward component of the air flow when the angle of attack of a wing is increased, the increase in the effective angle of attack for the duct entry is greater than the actual increase in the angle of attack of the wing. In fact, the effective angle of attack may increase one and one-half to two times as much as the angle of attack of the wing. On bombers, for which the difference in angle of attack for cruising and for climbing is small, the effect is probably not important. On pursuit airplanes, however, it may be important.

The safest procedure in the design of wing entrances is to investigate the entrances on the actual airplane in flight. In this way, the effect of the propeller is included. When the entrance is studied on a model in a wind tunnel, great care is necessary to avoid obtaining a misleading result. The effect of the entrance on the lift is most evident near the wing stall angle. The effect of the wind-tunnel walls may dominate the result near this angle and supporting strut interference may greatly exaggerate the losses caused by the entrance.

COOLING-AIR EXIT

The cooling-air exit should be designed to introduce the cooling air back into the air stream in such a manner that the drag will not be unduly large. For example, the low-energy cooling air may cause separation of the flow from a nacelle. Separation is likely to occur if the cooling air is emitted at a place where the flow is already near the normal limit of expansion; thus, it is much easier to make a low-drag exit on the bottom of a wing than on the rear top part of a wing.

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The air flow through a heat exchanger should be controlled by varying the size of the exit. In high-speed flight at low altitudes, the dynamic pressure of the free stream is high and the pressure drop available for cooling is greater than is necessary. Under these conditions, the area of the exit is reduced in order to reduce the flow of air and the pressure drop through the exchanger. In the condition of climb, the pressure drop required by the exchanger frequently equals or exceeds the dynamic pressure of the free stream. The exit area is then made large. That the exit area must undergo large changes to accommodate large changes in the volume rate of flow under various conditions of airplane speed, altitude, and rate of heat transfer is evident from an earlier discussion.

DRAG OF AN INSTALLATION

Form Drag

The form drag of an installation is largely a function of its frontal area. For instance, a completely submerged installation has no form drag except possibly that caused by a poor air entrance or exit. At the other extreme, a large nacelle on a thin wing has a relatively large form drag. In practice, nacelles are submerged in wings various amounts. For all these arrangements, the drag of the nacelle is very nearly proportional to the frontal area of the nonsubmerged part.

Streamline bodies and nacelles with conventional NACA cowlings have been tested many times to determine the relative form drag of these bodies. The results of many of these tests are summarized in figure 41, in which drag coefficient based on frontal area is shown as a function of fineness ratio. The dashed line is for streamline bodies of revolution; the solid line is for the same bodies equipped with conventional NACA cowlings. Fineness ratio is defined here as the ratio of the length behind the maximum section to the diameter of the maximum section. It will be noted that the bodies equipped with nose cowlings are much more sensitive to the treatment of the afterbody than the bodies with rounded noses. This fact shows that the conventional cowling introduces some sort of instability into the flow which may cause separation and result in a high form drag. This instability can be counteracted to a large extent by alining the afterbody with the flow. In the setup being considered, alinement of the afterbody means the use of long afterbodies in order that the rate of expansion of the flow over the afterbody is low.

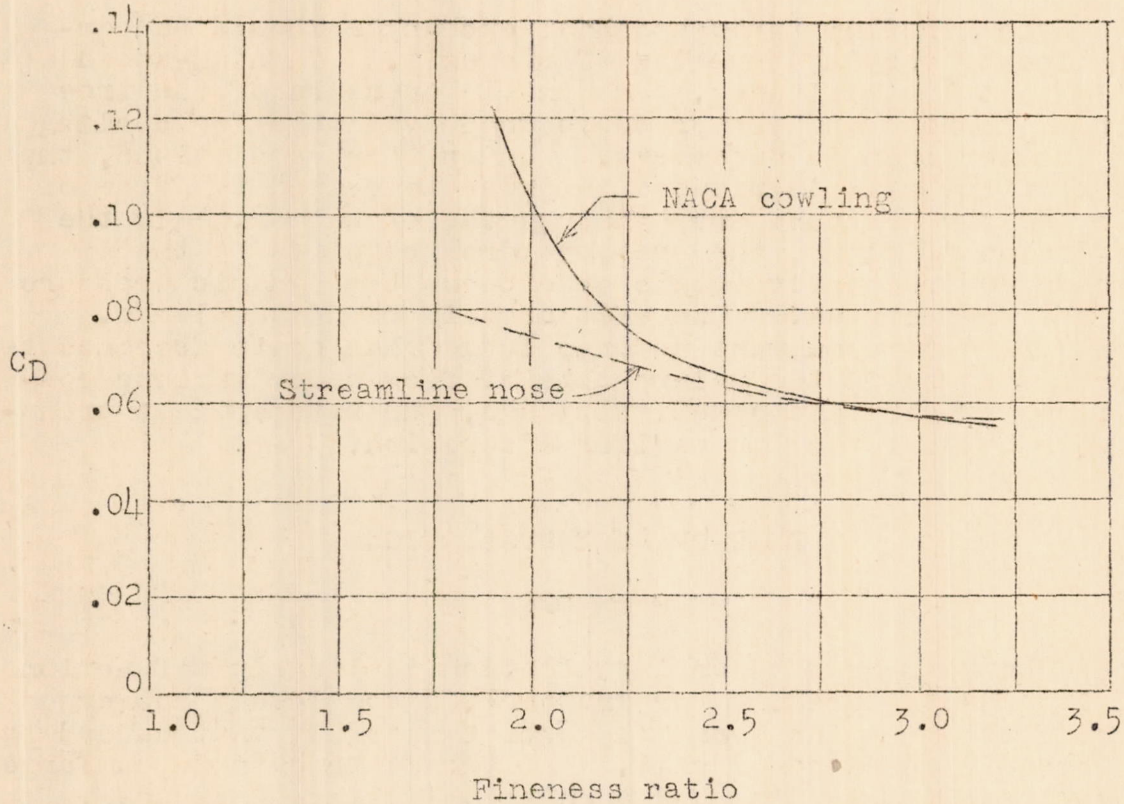

Figure 41. - Effect of fineness ratio on drag coefficient of body of revolution with streamline nose and with NACA cowling nose.

The situation is similar when the nacelle is installed on a wing. If the nacelle is not alined with the flow over the wing, the unstable flow causes separation on the nacelle and the wing and a high drag results. Exactly true alinement obviously cannot be obtained except at one angle of attack. The alinement should therefore be made for the condition of flight in which the best performance is desired.

Analyses and wind-tunnel tests show that engine cooling can be obtained for about 5 percent of the engine power, which is a relatively low cost; the exact amount depends upon the critical altitude of the engine. The cost of engine cooling, however, in almost all actual installations is much larger than 5 percent of the engine power. For a critical altitude of 25,000 feet, for example, an installation that gives engine cooling at a cost of only 10 percent of the engine power is considered particularly good. The high cost generally can be largely attributed to poor design of the afterbody of the engine cowling.

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Cooling-Air Drag

The drag of an installation other than the form drag previously discussed is caused by the return to the air stream, through the exit, of air that has less momentum than the free stream. Consider the flow of air through a heat-exchanger installation. In the entrance, the total pressure of the air is essentially that of the free stream H_0. In the heat exchanger, the air does useful work and suffers a corresponding loss in total pressure. In the ducts, non-useful work is done and further pressure is lost. Finally, at the exit the total pressure H_3 is given by

$$H_3 = H_0 - \Delta H$$

This total pressure H_3 is composed of static pressure p_3 and dynamic pressure. The cross section of the flow at the exit is such that the static pressure in the exit equals the static pressure outside the exit. The velocity of flow out of the exit is therefore determined by H_3 and p_3. The total pressure H_3 generally is small enough for the velocity of the flow out of the exit to be less than the free-stream velocity V_0. The system must then be charged with a drag that is given by

$$D = \frac{W}{g} (V_0 - V_3)$$

PROVISION OF PRESSURE DROP

Available Pressure Drop as a Function of

Free-Stream Dynamic Pressure

Providing adequate pressure drop is one of the functions of an installation. A good entrance headed into the air stream provides a total pressure that is very nearly equal to the total pressure of the free stream. The drop in total pressure that is available for use in a heat exchanger and the accompanying ducts and diffusers is the difference between the total pressures in the entrance and in the exit. In the limit, that is, when the exit area is very large, the total pressure in the exit consists solely of static pressure and the exit velocity is zero. The available total pressure then is the difference between the total pressure in the entrance and the static pressure in the exit. The static pressure in the exit depends on its location. If the exit opens to the rear and the velocity of the air over the outside of the exit is nearly free-stream velocity, the

static pressure in the exit is very nearly free-stream static pressure. The pressure drop available for the installation is then the free-stream dynamic pressure.

The assumption that the pressure drop available for use in any installation is equal to the free-stream dynamic pressure usually is not true in practice. When the airplane is motionless on the ground, the free-stream velocity is zero; yet a pressure drop is available because the propeller produces a flow over the entrance and the exit. In flight, also, the propeller has an effect. The total pressure in an entrance that is located behind the inner sections of a propeller may be less than the total pressure of the free stream on account of the blocking effect of the propeller shanks. The total pressure in an entrance that is located behind the outer sections of a propeller may be greater than the total pressure of the free stream. If the exit is located in the slipstream, the static pressure in the exit likewise will be less than that of the free stream. Furthermore, flaps can be used to reduce the pressure at the exit and to increase the pressure drop available for cooling. Under favorable conditions, flaps can increase the available drop by some 30 percent. It must be realized that the radial location of both the entrance and the exit with respect to the propeller and the distance of the entrance and the exit behind the propeller determine the pressure drop that is available.

Pressure Provided by Conventional Cowlings

Alinement of the entrance with the local air flow is an important consideration. An airplane is required to operate through a range of angle of attack and this requirement handicaps the performance of the entrance. Even a total-head tube, which is much like a cowling or scoop, has a limited angle of attack through which approximately the full total-pressure reading is given. Entrances on airplanes are usually more critical to change in angle of attack than total-head tubes. At positive angles of attack, crossflow over an entrance allows air to enter at the bottom and to flow out the top. The usual name for this phenomenon is spillage. (See fig. 42.)

Conventional NACA cowlings experience spillage and engines installed in such cowlings tend to have the lower cylinders overcooled and the upper ones undercooled. Numerous attempts have been made to eliminate spillage. The most successful scheme is the use of a high-speed entrance.

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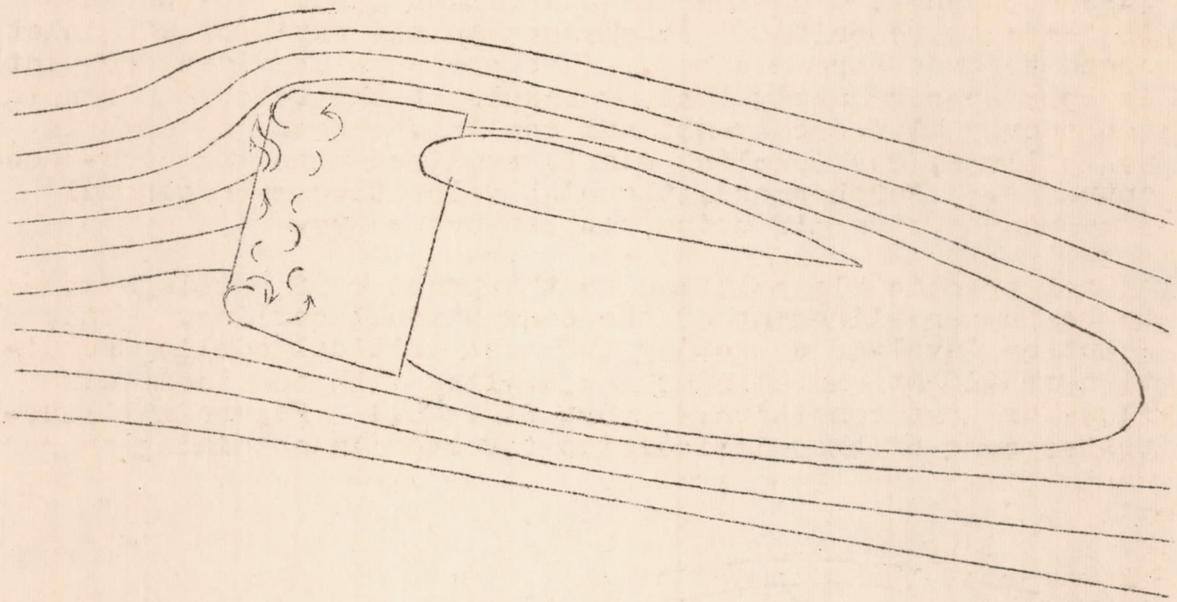

Figure 42. - Spillage from poorly alined NACA cowling.

This type of entrance has a comparatively small inlet area, and the air is inducted at high speed compared with the low entrance speed in the case of the conventional cowling. (See fig. 43.) The high-speed entrance tends to give a

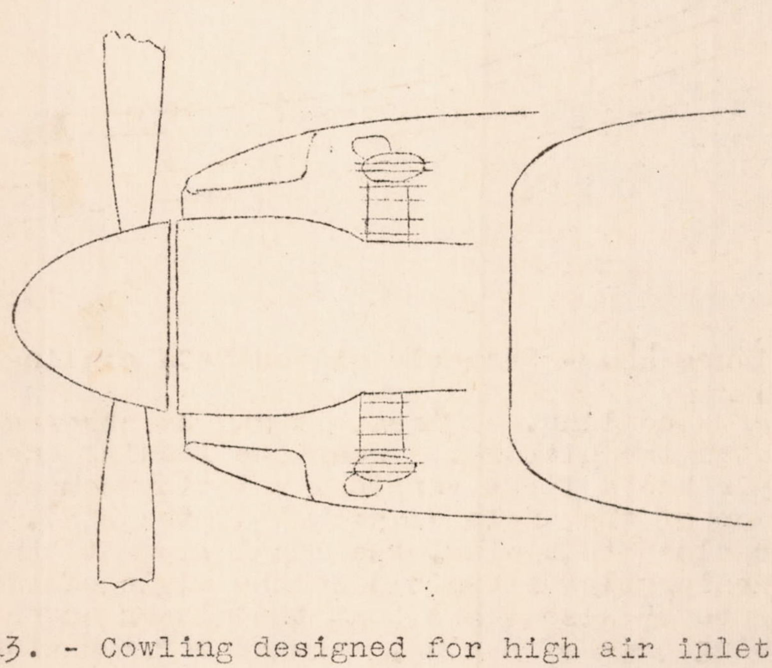

Figure 43. - Cowling designed for high air inlet speed.

uniform pressure across the engine. This entrance, however, does not function well under all flight conditions because it tends to be critical to changes in the ratio of air inlet speed to free-stream speed. The pressure provided frequently is only approximately that available at the top cylinders in the poorly alined conventional cowling. Because there is no spillage, more cooling air is required with the high-speed entrance. Furthermore, the high velocities make careful treatment of the diffusing section necessary.

A practicable solution to the problem of spillage appears to be proper alinement of the conventional cowling. This solution involves selecting the most critical cooling condition of flight and alining the cowling with the local air flow for that condition. (See fig. 44.) Figure 45 illustrates some of the difficulties involved in obtaining proper

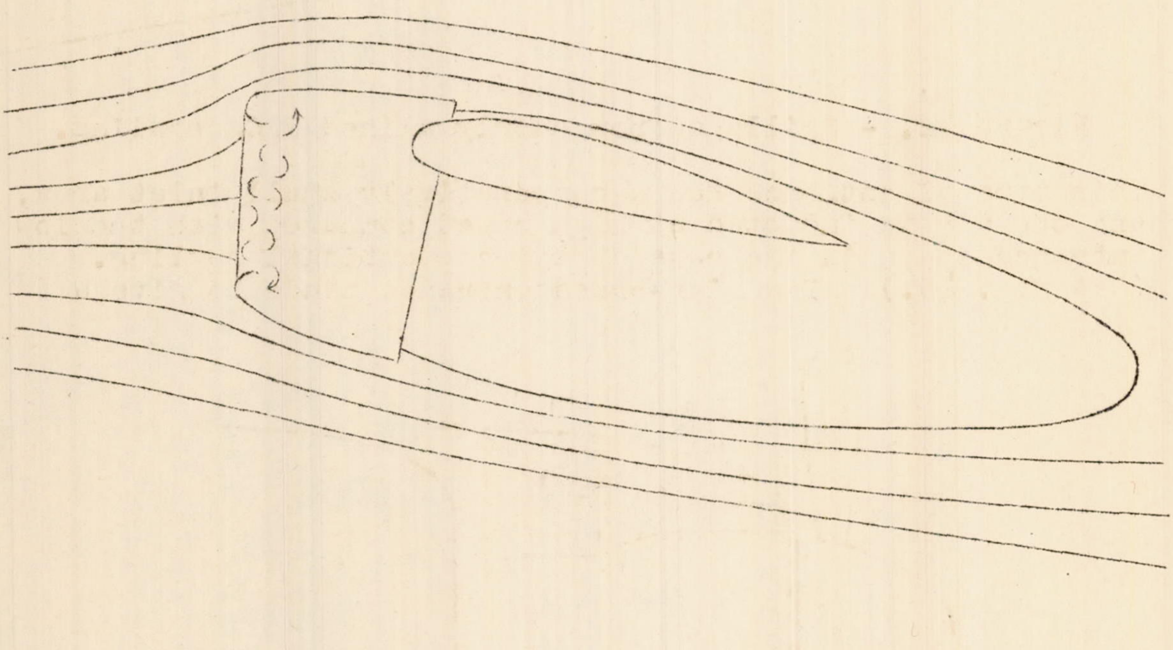

Figure 44. - Properly alined NACA cowling.

alinement of a cowling. First, it may be observed that the wing dominates the picture. Near the leading edge of a wing, the air has a large vertical velocity component that is a function of the angle of attack of the wing. If an entrance is close to a wing, the local flow at the entrance is therefore largely determined by the angle of attack of the wing. Figure 45 also shows that the closer an entrance is to the leading edge of a wing, the greater the change in the direction of the local flow with change in the angle of attack.

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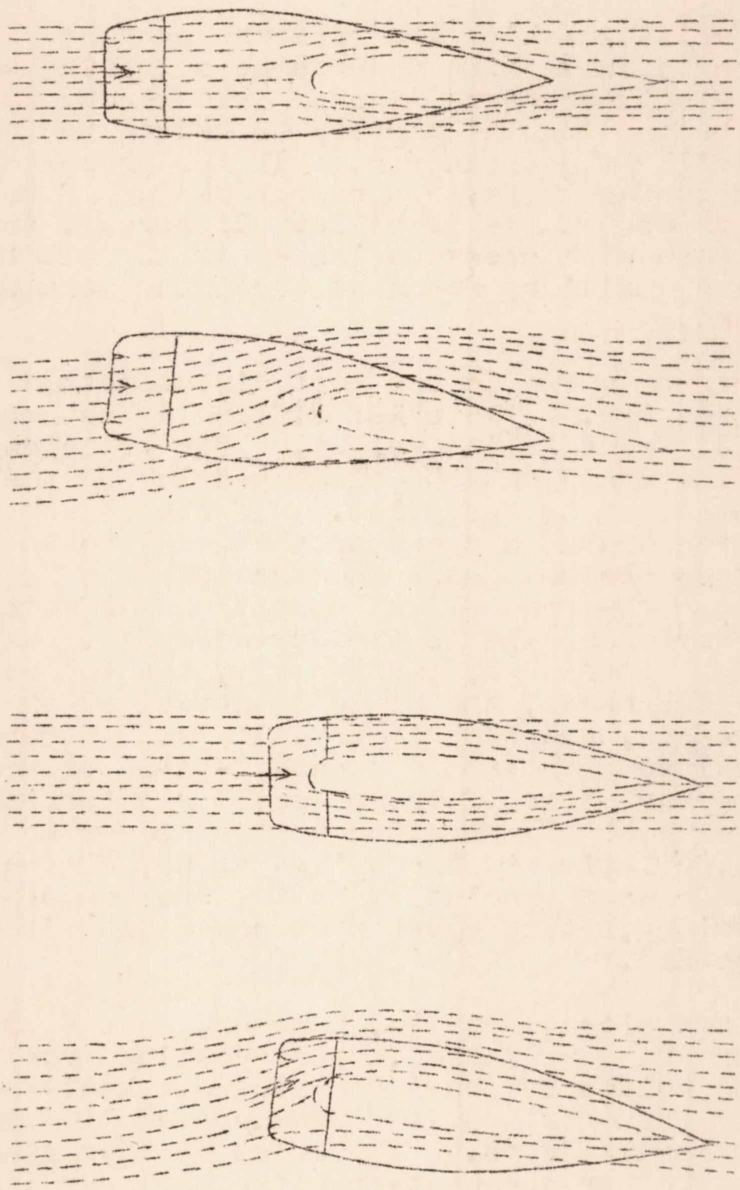

Figure 45. - Influence of wing on air-flow direction at cowling entrance.

Augmentation of Pressure by Blowers

Cases arise in which the pressure provided by the normal installation is inadequate under some operating conditions. The most logical procedure in such cases is to use more surface area for cooling and thereby to reduce the pressure drop required by the exchanger. If the greatest possible increase in surface area, the proper design of entrance and exit, maximum use of the propeller slipstream, and the reduction of nonuseful pressure losses still fail to solve the problem, some auxiliary means of providing more pressure must be found.

A blower or a fan is the most direct method of providing increased pressure. The chief difficulty connected with the use of a blower is its weight. For efficient blower operation, the speed and the blade setting of the blower must be controlled by the pilot. If the blower operates at constant pitch and at a speed proportional to the engine speed, it does the most work and provides the highest pressures in the high-speed flight condition for which the cooling pressure usually is already adequate. As the altitude of flight increases, furthermore, the velocity of the cooling air increases, the angle of attack of the blower blades decreases, and the blower unloads. Figure 46 shows for a typical constant-speed constant-pitch blower, for example, the variation of the pressure rise across the blower and the power cost of operation of the blower as functions of altitude, relative to the values at 50,000 feet. If a blower is used as the solution of the problem of high-altitude cooling, it is clear that speed and pitch controls must be provided.

Speed and pitch controls, however, make a blower excessively heavy and complicated. All cooling problems can doubtless be solved by providing adequate heat-transfer surface area for obtaining the required cooling with the available pressure drops. The question that remains to be answered is whether the heat-transfer engineer will choose to provide the required surface area or whether he will decide to use a blower with the associated weight and complication.

MECHANISM OF COOLING IN AIR-COOLED ENGINE

In the following discussion, the aim is to describe some of the details of the mechanism of the cooling of an air-cooled engine, in order to give an insight into the phenomena

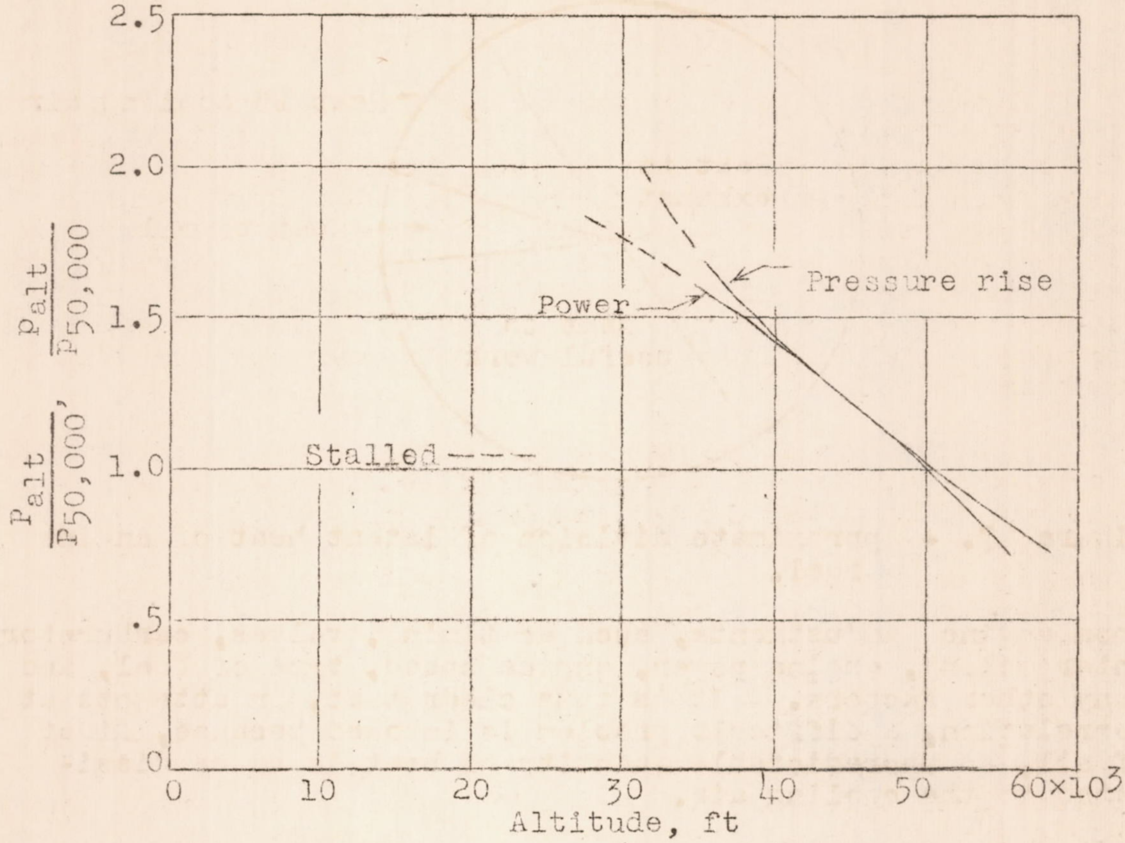

Figure 46. - Effect of altitude on power cost and pressure rise of constant-speed constant-pitch blower.

that are involved, and to point out the numerous factors that must be taken into account in trying to obtain correlations and the inherent difficulties that are encountered in making measurements.

The mechanism of heat transfer for such cooling units as radiators and intercoolers is well known. Here the pressure drop for cooling, the rate of heat transfer, the flow of cooling air, the flow of fluid to be cooled, and the temperatures can all be fitted into a consistent picture and correlated with one another. The conditions encountered in the air-cooled engine are complicated by numerous factors that make such a correlation extremely difficult.

In the air-cooled engine, the latent heat of the fuel is divided in some such proportion as that shown in figure 47.

The division of the heat pictured here is only a very rough approximation because the relative division depends

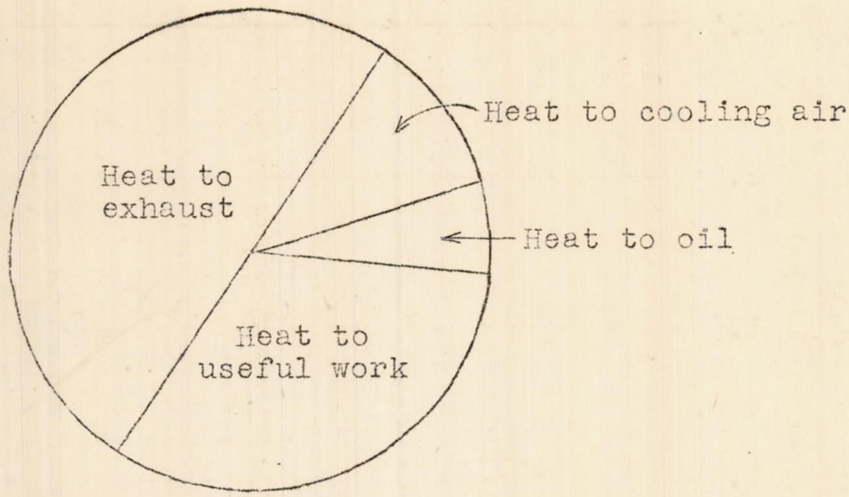

Figure 47. - Approximate division of latent heat of engine fuel.

upon engine adjustments, such as timing, valves, carburetor, intercooling, engine power, engine speed, type of fuel, and many other factors. It is thus clear that, in attempts at correlation, a difficult problem is imposed because, first of all, an unpredictable quantity of heat is to be dissipated to the cooling air.

The over-all problem of obtaining a correlation between engine temperatures and pressure drop is further complicated by the mechanism of cooling. The heat going into cooling is divided somewhat as shown in figure 48. It is interesting

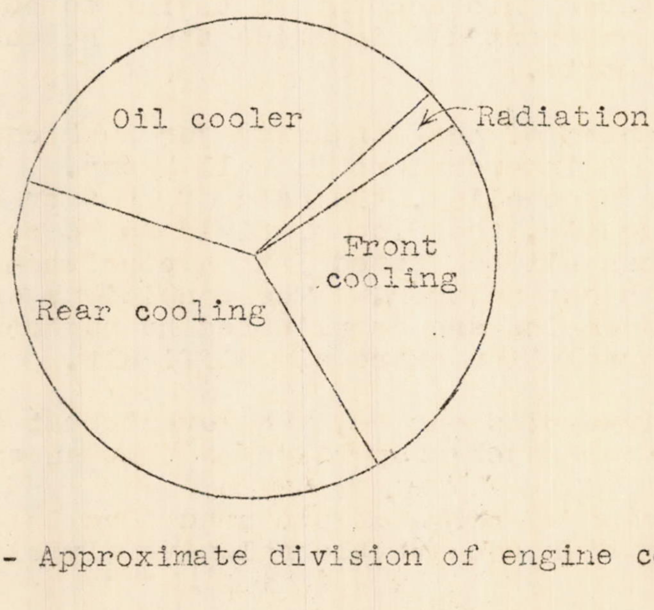

Figure 48. - Approximate division of engine cooling.

to observe that only approximately 40 percent of the heat is dissipated by the baffled part of the cylinder. Only about 40 percent of the cooling, therefore, can be directly correlated with the pressure drop of the engine cooling air.

It is clear that the heat to be dissipated to the cooling air is a complex function of many factors and that the cooling is only partly dependent upon the pressure drop. The nature of the cooling-air flow, the means of obtaining pressure, and the method of measuring pressure on an air-cooled engine will be discussed in the following sections.

Front Pressure

The flow in front of the engine takes numerous forms, depending upon the conditions of flight, the altitude of the airplane, and the type of cowling. For instance, consider the air flow in a cowling in the high-speed flight condition. In this case, the cowling operates very much like a total-pressure tube and a measurement of the total pressure in the entrance shows very nearly free-stream pressure there.

In the take-off or climbing condition, the propeller superposes a swirl on the axial flow in front of the engine (fig. 49). The velocity pressure, the total pressure, and

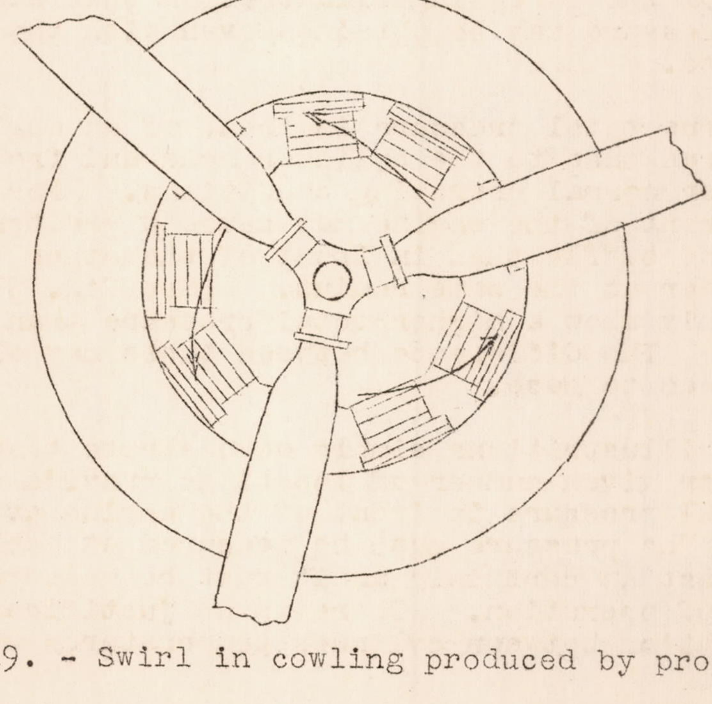

Figure 49. - Swirl in cowling produced by propeller.

the static pressure then vary radially across the front of the engine.

A spinner is often used to cover the propeller hub and, in this case, a relatively high-speed jet is also present (fig. 50). The pressure tubes must be carefully located to obtain the true pressure.

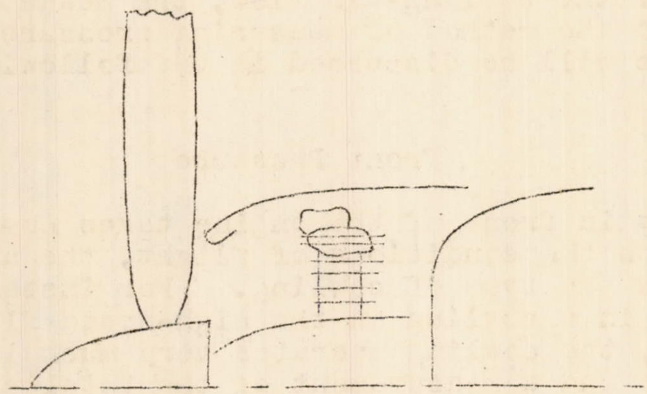

Figure 50. - High-speed jet in small entrance area.

Under flight conditions, the cowling is operated through a range of angle of attack. When the cowling is operated at a high angle of attack as in the climbing condition, the air tends to flow into the bottom of the cowling and out the top and spillage occurs. (See fig. 42.) It may easily be deduced from the foregoing illustration that wide variations in total pressure may be obtained even with the most careful measurements.

The true total pressure in front of an engine varies from the crankcase to the cylinder head and from top to bottom under normal operating conditions. For instance, the swirl in front of the engine may cause a greater pressure in front of one baffle than in front of the other baffle on the same cylinder at the same radius. (See fig. 51.) Tube A will probably show a higher total pressure than tube B (fig. 51). The difference between tubes may also vary from cylinder head to base.

These illustrations simply demonstrate that no single location nor given number of locations provide a true picture of the total pressure in front of the engine available for cooling. The pressure must be measured at each point of interest just as certainly as it must be measured for each condition of operation. There is no justification, otherwise, for correlation between cylinder temperatures and pressure

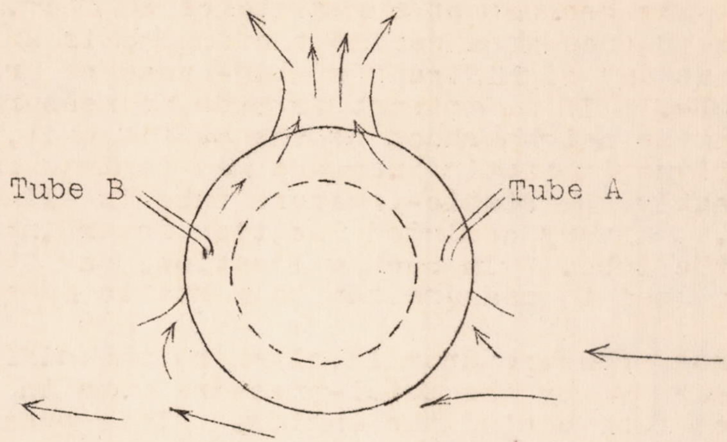

Figure 51. - Flow at baffle entrance and exit.

drops across cylinders. There is no more justification for expecting one pressure measurement on a cylinder to be significant than for expecting one temperature measurement to be significant.

The individual tubes must be placed in the fins or in the baffles where the value of the pressure is desired. The tube preferably should be installed in the fins a short distance behind the entrance to the baffle. (See fig. 51.) A tube so located gives the true total pressure at the given location.

Rear Pressure

The flow behind the engine presents a much less complicated picture than the flow in front of the engine. Here the chief complication arises from the jets of air as the flow escapes at the baffle exits. (See fig. 51.)

If the static pressure is measured behind the engine, it will be found constant unless the measurements are made in the neighborhood of the baffle exit or near the cowling exit where the air is accelerating.

The total pressure of the flow out the exit slot might be measured but such a measurement with flaps and adjustable cowling exit may offer some complications. A static-pressure measurement at a point remote from the baffle exits and the cowling exit is most convenient. The true static pressure must be constant behind the engine except near the exit slot. A static-pressure gradient could occur behind

the engine only because of a resistance to flow. A simple calculation of free area behind the engine is sufficient evidence that any significant static-pressure gradient would be impossible. If an attempt is made to measure the static pressure in the neighborhood of the baffle exit, however, wide variations in static pressure may be measured, depending upon how nearly the static-pressure tube is alined with the local flow. A very secluded location accordingly should be found for the tube. In such a location, any type of measuring tube may be used to measure the true static pressure.

The true pressure drop is given by the difference in pressure measured by the total-pressure tube in front of and the secluded tube behind the engine. This pressure drop applies to the passage in which the total-pressure tube is located and, under different conditions, may vary from point to point over the engine.

Effect of Propeller

Some years ago a series of tests was run on a family of cowlings with and without propeller operating. Some of the tests were made with an electrically heated cylinder mounted on a crankcase and some of them were made on an actual operating engine.

The case of the electrically heated cylinder demonstrates some of the cooling problems without the complications of variations in heat dissipation due to engine operating conditions. Figure 52, taken from figure 11 of reference 22, shows the effect of a propeller and a blower on the temperature distribution around an electrically heated cylinder. When the results are plotted as in figure 53, which is taken from figure 13 of reference 22, it may be seen that, even over the baffled part of the cylinder, the temperatures are not determined solely by the pressure drop.

The fact that figure 53(a) shows correlation even for the front of the cylinder simply means that the disturbances in the flow which cool the front of the cylinder are related to the speed of the air. The baffled part of the cylinder has a flow and a pressure drop that are related to the speed of the airplane.

Effect of Cowling-Entrance Size

The measurements described were all made with a given cowling except where the blower was used. The next step -

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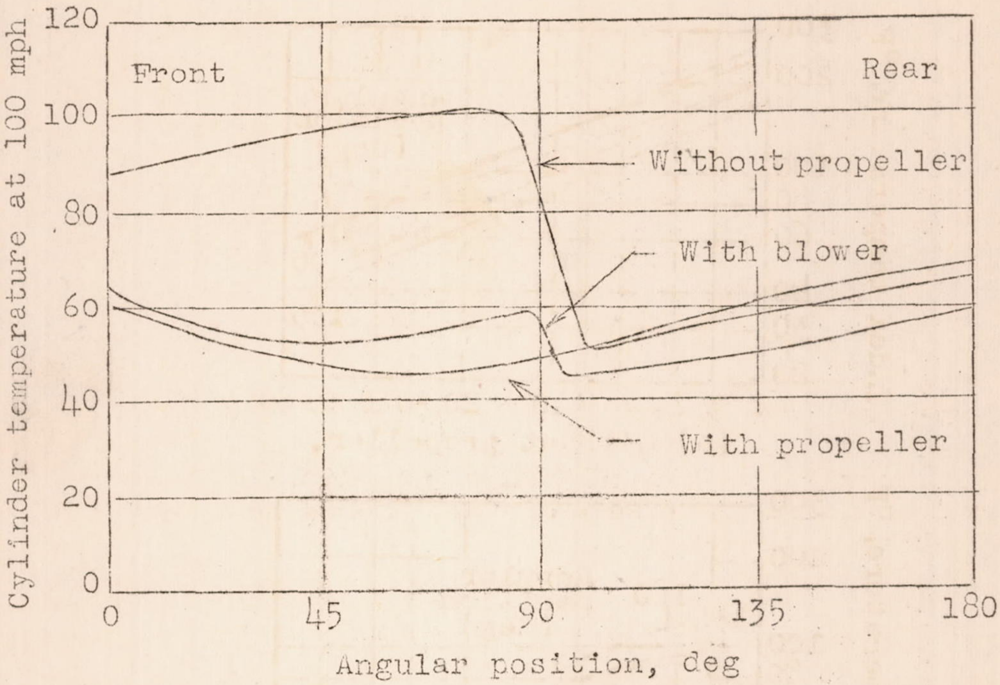

Figure 52. - Temperature variation around an electrically heated cylinder for various conditions. (Data from fig. 11 of reference 22.)

what happens to the cooling in an actual installation - is also interesting. Figure 54 illustrates the effect of open frontal area on the heat dissipated to the cooling air as measured from the air flow and the temperature rise of the air. No startling conclusions may be drawn from the curves of figure 54. It may be noted, however, that the cowlings with the largest openings dissipate the least amount of heat to the cooling air. This trend may be explained by spillage, whereby part of the heat transferred to the air in front of the engine is spilled out the front of the cowling.

The tendency of the smaller cowling opening to dissipate more heat to the cooling air in the baffles or to require a higher pressure to maintain a constant cylinder temperature is also experienced on the test stand. Engines tested for cooling on a test stand where the engine is installed in a circular duct and all the cooling air passes over the engine invariably require higher pressure drops for cooling than the same engines installed in conventional cowlings on aircraft.

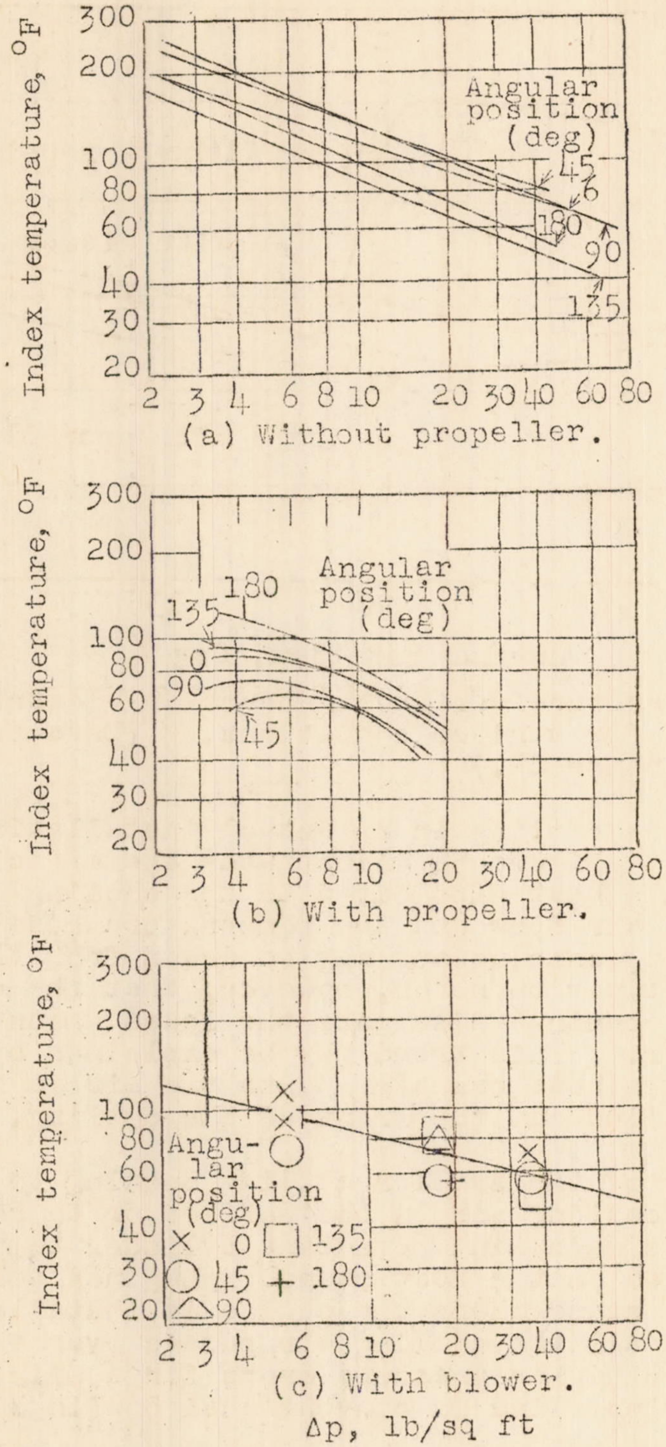

Figure 53. - Cylinder temperature as a function of pressure drop for various conditions. (From fig. 13 of reference 22.)

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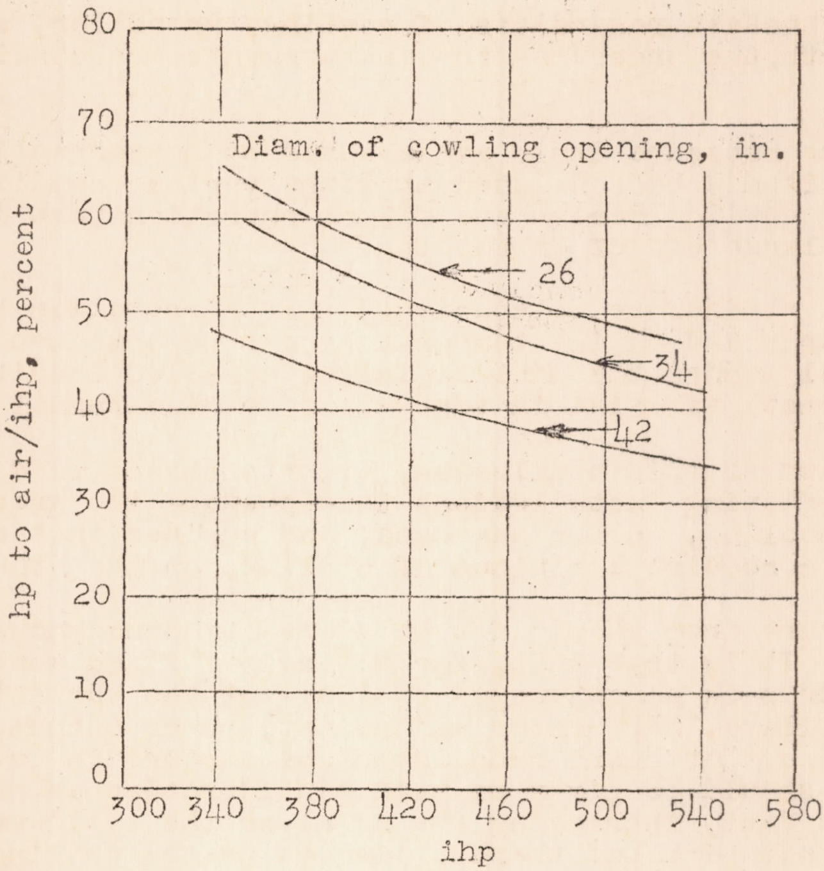

Figure 54. - The ratio of horsepower transferred to cooling air to indicated horsepower as a function of indicated horsepower for various cowling openings.

Conclusions

The foregoing discussion and illustrations simply demonstrate that the pressure drop required for cooling an engine is a function of many factors, all of which vary for each test setup and each flight condition. The relative effect of these variables may change when flight, wind-tunnel, and test-stand results are compared.

Throughout the tests just discussed, elaborate measurements were taken of the pressures and the velocities at numerous points in front of and behind the engine, in the fins on the front of the cylinder and under the baffle, and at the exit to the cowling. Such measurements establish an effective leak area representing the leak area for cooling air around the engine. Because this effective leak area

remained constant regardless of cowling, propeller, spillage, and so forth, the pressure-drop measurements are considered reliable.

If the pressure drops are assumed to be correctly measured, it may be concluded that the various conditions imposed by cowling design and flight operation have a surprisingly large effect on cooling.

It may therefore be concluded that correlation between pressure drop and engine temperature can be obtained at present only under the most rigidly controlled conditions, such as sometimes exist in test-stand or wind-tunnel tests.

In view of this conclusion, efforts should be directed toward developing installations that produce the greatest possible cooling, on the one hand, and engines that are adjusted to require a minimum of cooling, on the other hand.

Pressure drop should not be viewed as synonymous with cooling. It is true that, for any set of fixed conditions, pressure drop is a measure of cooling but the variation of swirl, spillage, carburetor adjustment, power output, engine speed, and so forth are such important factors in cooling that they should be given as much consideration as pressure drop. In fact, increasing the swirl or the spillage in front of an engine and thereby increasing the cooling may be more feasible in some cases than obtaining the same increase by increased pressure drop in the baffles.

DUCTS

Means and methods of reducing the internal losses of total pressure that occur in ducts are considered in the present section. The experimental data used are taken principally from references 23 to 25. Total-pressure losses occur in ducts because of (1) friction, (2) obstructions, (3) change of shape, (4) bends, (5) abrupt contractions and expansions, and (6) diffusers. The loss in any part of a duct - for example, at a corner or in a diffuser - for all practical purposes may be taken as proportional to the dynamic pressure at that part. The total loss of pressure in a duct may then be written

$$\Delta H = \Sigma kq$$

If a duct can be designed large enough that the dynamic pressure in the duct is small, the loss in total pressure caused by bends, and so forth, obviously is small, inasmuch as k is roughly of the order of unity. It is not therefore necessary to exercise much care in trying to design for small values of k, and the following discussion can be largely disregarded. If, on the other hand, it is necessary for the dynamic pressure in a duct to be large, losses can be reduced by making the values of k small.

Friction

The loss due to friction in ducts is generally negligibly small. It can be calculated by the equations given in the section entitled "Pressure Loss."

Obstructions

Obstructions (pipes, spars, and so forth) should be eliminated insofar as possible. When the obstructions are small compared with the size of the duct, the usual fairings may be used. When the obstructions are large, the fairings should be based on the rules for good diffuser design, in order to give a low rate of expansion following the obstruction.

Change of Shape

A gradual change of duct shape at constant cross-sectional area (for example, a change from circular to square section in a distance at least equal to the diameter of the circle) gives a negligible loss.

Bends

The loss coefficient $k = \frac{\Delta H}{q}$ for a bend is a function of

(1) The radius ratio or the ratio of the radius of curvature R of the duct axis to the diameter or width of the duct D measured in the same plane as R. (See fig. 55.)

(2) The aspect ratio or the ratio of duct height W to width D.

(3) The angle θ through which the air is deflected by the corner.

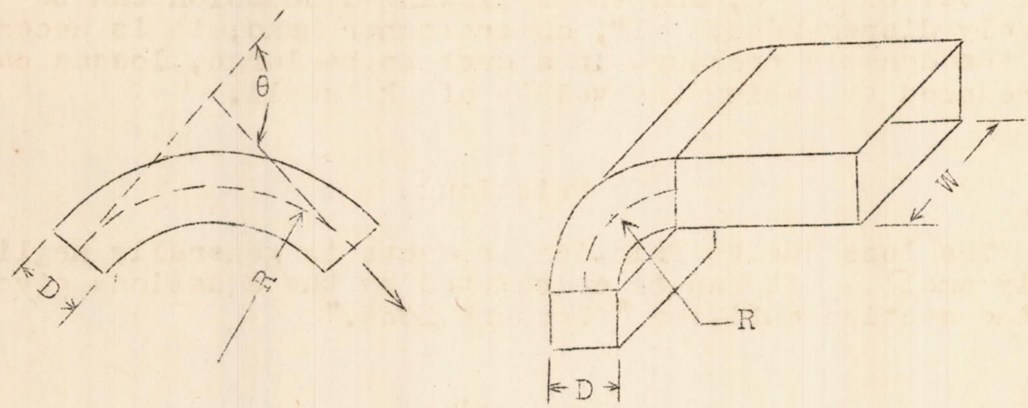

Figure 55. - Definition of duct symbols. (From fig. 1 in reference 25.)

Figure 56 shows the effect on k of varying R/D and W/D for a bend of 90°. It also shows that a circular section is better than a square section and that a rectangular section with the turn made on the short side (W > D) can have a smaller k than a circular section.

If it is not practicable to design a bend with desirable (high) values of aspect ratio and radius ratio, vanes should be used to reduce losses. As the gap-chord ratio s/c of the vanes is decreased, the aspect ratio and the radius ratio of the bend are increased. The additional vanes, however, increase the frictional loss. It has been found that, for thin metal vanes with the shape of the arc of a quarter circle (fig. 57), the total loss is a minimum (k = 0.2) at about s/c = 0.45.

Bend losses are lower when the bend is followed by a straight duct than when the duct is terminated immediately after the bend. For θ = 90°, the straight part should be at least 4D in length.

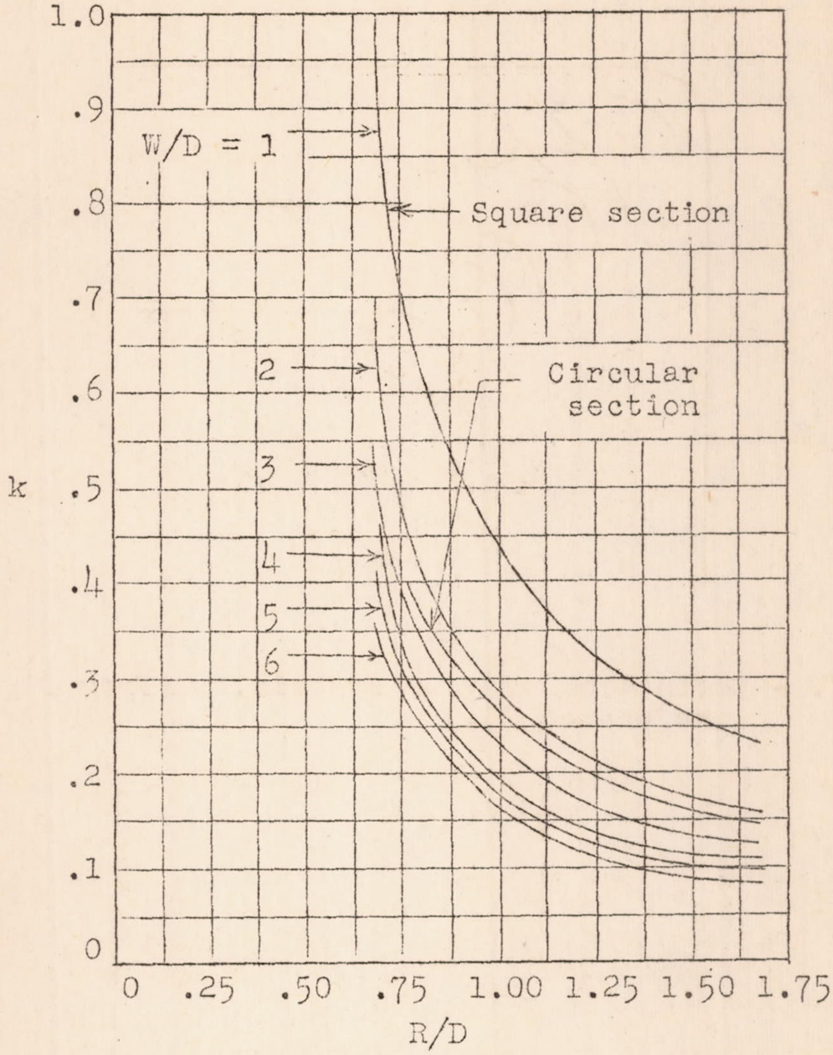

Figure 56. - Loss coefficient k of bend as a function of ratio of height to diameter and ratio of radius of curvature to diameter. Square, circular, and rectangular cross sections; $\theta = 90^{\circ}$. (From fig. 3 of reference 25.)

Abrupt Contractions and Expansions

Losses in the total pressure of the fluid occur at abrupt changes in cross-sectional area. The losses that take place at abrupt contractions of cross-sectional area (fig. 58) cannot be calculated without the aid of experimental data. Such data for incompressible fluids can be correlated by the equation (p. 737, reference 26)

$$\Delta H = kq_2$$

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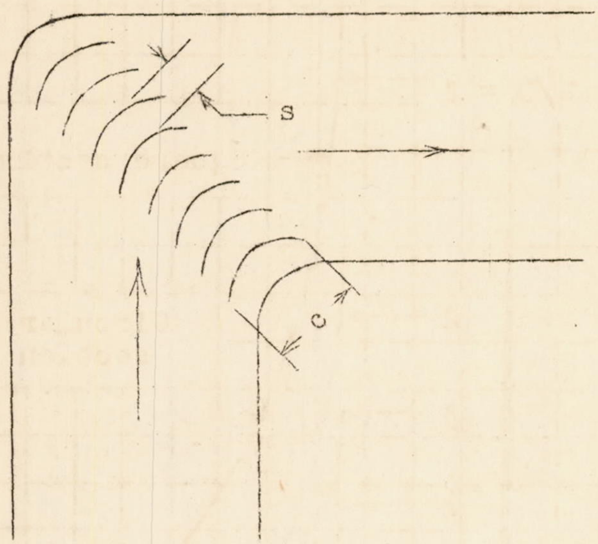

Figure 57. - Definition of vane symbols. (From fig. 5 of reference 25.)

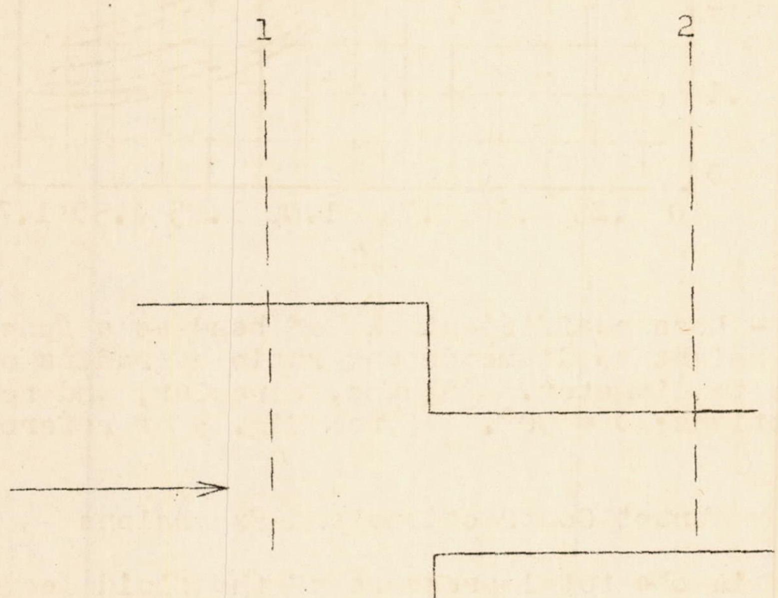

Figure 58. - Abrupt contraction of cross-sectional area.

where

ΔH loss in total pressure

q_2 dynamic pressure in smaller cross section

$k = 0.4\ (1.25 - f), \qquad f < 0.715$

$k = 0.75\ (1 - f), \qquad f > 0.715$

f ratio of area of smaller cross section to area of larger cross section

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The losses at an abrupt contraction can be materially reduced by rounding the edge at the entrance to the smaller duct. If the edge is only slightly rounded - as, for example, by a few strokes of a file - the loss is reduced by approximately one-fourth.

An abrupt increase in the cross-sectional area of a duct (fig. 59) also causes a loss in the total pressure of the fluid. This case is one of the few in which loss of pressure in flow through ducts can be calculated without the aid of experimentally determined coefficients. Most

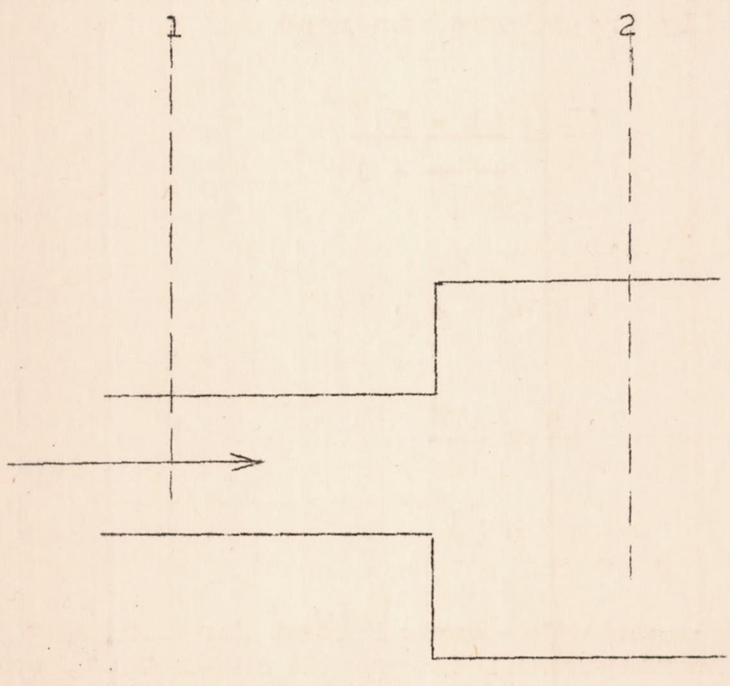

Figure 59. - Abrupt expansion of cross-sectional area.

textbooks on hydraulics - for example, reference 27 - show that, for incompressible fluids, the loss in total pressure caused by a sudden expansion is given by the equation

$$\Delta H = (1 - f)^2 q_1 \tag{68}$$

Here again the dynamic pressure is to be taken at the smaller section.

For compressible fluids, the loss in total pressure caused by a sudden increase in cross-sectional area, as derived in an unpublished analysis, is given by the equation

$$\frac{\Delta H}{H_1} = 1 - \left(\frac{p_2}{p_1}\right)^{\frac{1}{1-\gamma}} \left(\frac{1}{f} - \frac{\frac{p_2}{p_1} - 1}{\gamma f^2 M_1{}^2}\right)^{\frac{\gamma}{1-\gamma}} \tag{69}$$

where

$$\frac{p_2}{p_1} = \frac{1 + \gamma f M_1{}^2 + \gamma\sqrt{2\gamma f M_1{}^2 (1 - f) + 1 - 2f^2 M_1{}^2 + f^2 M_1{}^4}}{\gamma + 1}$$

For comparison of the losses in incompressible and in compressible flow, the equation for incompressible flow (equation (68)) is easily transformed to read

$$\frac{\Delta H}{H_1} = \frac{(1 - f)^2}{\frac{2}{\gamma M_1{}^2} + 1} \tag{70}$$

by using

$$H = p + q$$

and

$$M^2 = \frac{\rho V^2}{\gamma p}$$

$$= \frac{2q}{\gamma p}$$

Equations (69) and (70) are plotted for air, $\gamma = 1.405$, in figure 60. Inspection of figure 60 shows that, except at the highest Mach numbers, the loss for compressible flow can be considered the same for most purposes as the loss for

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incompressible flow and can therefore be calculated by the simple equation (68).

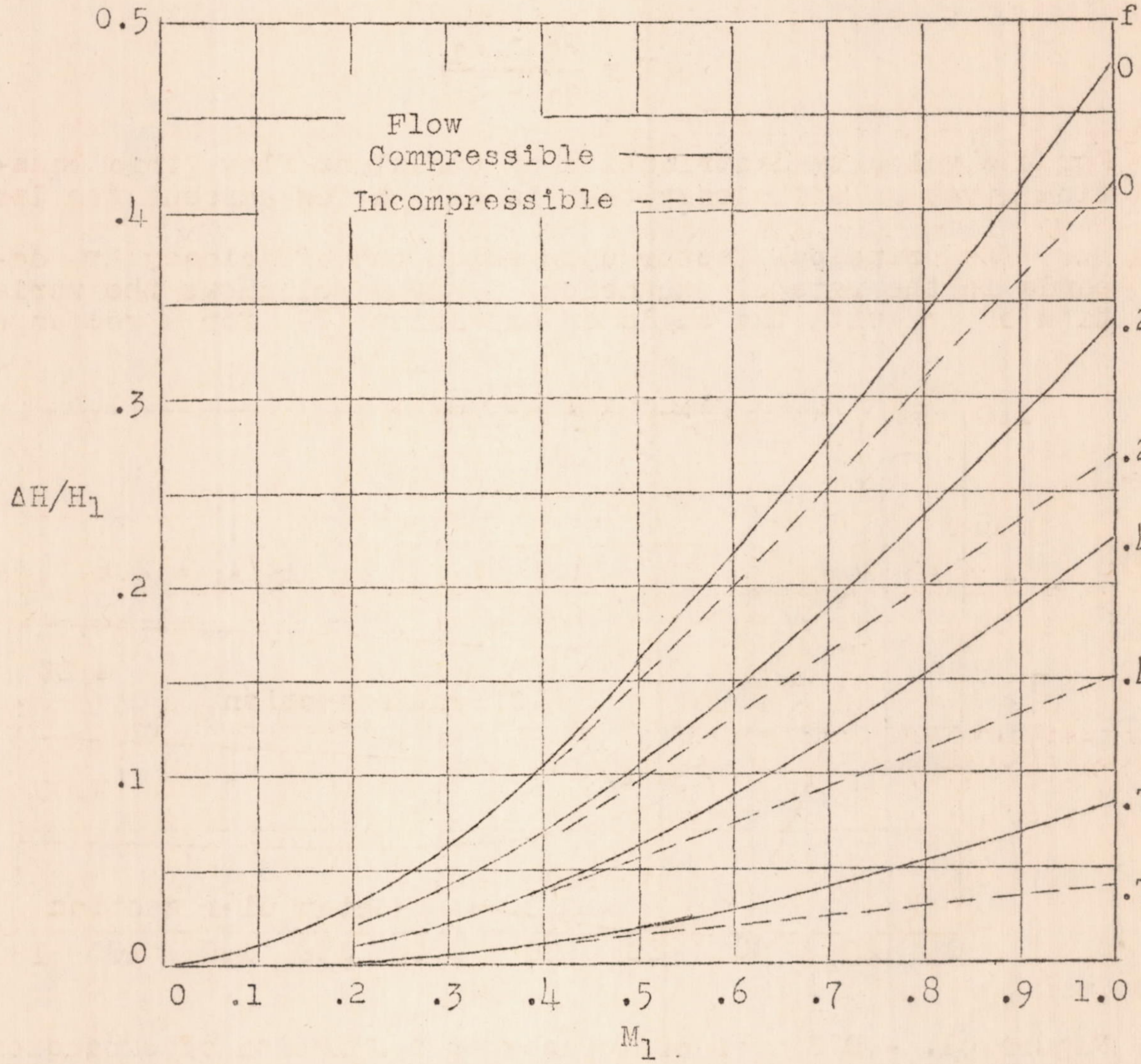

Figure 60. - Loss in total pressure at abrupt expansion as a function of area ratio and Mach number in compressible and incompressible flow. $\gamma = 1.405$.

Diffusers

The principal function of a diffuser, or expanding section, is to increase static pressure by transforming dynamic pressure to static pressure. The adverse static-pressure gradient in a diffuser tends to produce separation of flow at the walls and the separation causes a loss in total pressure. The principal design problem is to prevent separation.

If the distribution of velocity is constant across the initial and the final sections of a diffuser, the diffuser efficiency can be defined as

$$\eta = \frac{p_2 - p_1}{q_1 - q_2}$$

For the velocity distribution of turbulent flow, this equation gives an efficiency that is only a few percent too large.

The principal factor upon which the efficiency η depends is the rate of expansion. Figure 61 shows the variation of η with the angle of expansion 2θ for a rectangular

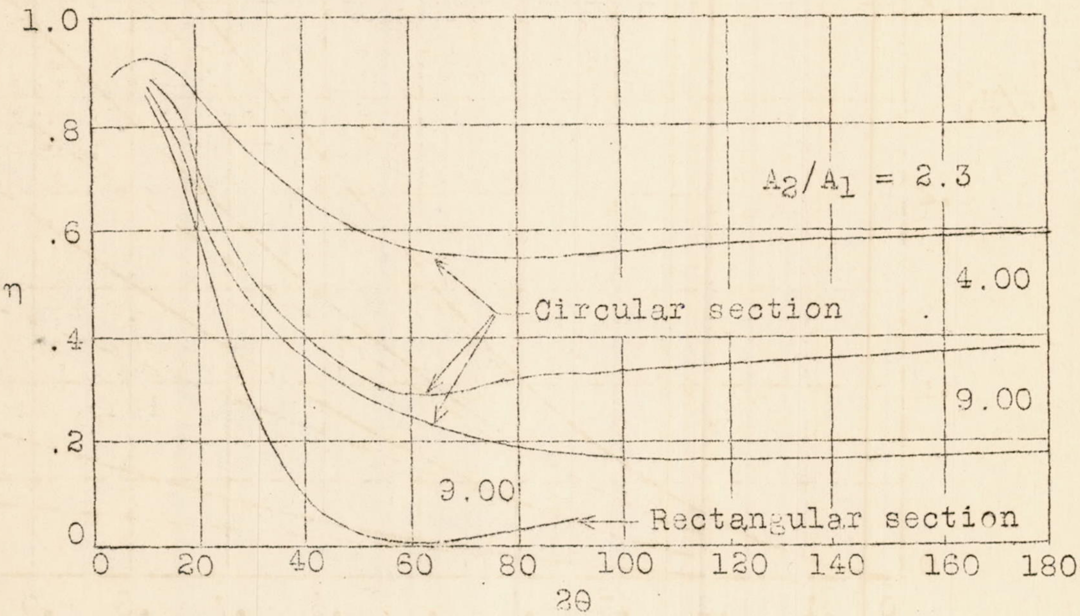

Figure 61. - Diffuser efficiency as a function of expansion angle and expansion ratio.

section and for circular sections of various expansion ratios. For conical diffusers, the angle of expansion of greatest efficiency lies between 5° and 8°. The efficiency decreases rapidly as 2θ increases beyond about 10°. For square diffusers, the angle of expansion for which the efficiency is greatest is about 6°. For rectangular diffusers in which the initial section is square and one pair of walls remains parallel whereas the other pair diverges, maximum efficiency occurs at about $2\theta = 11^\circ$. Figure 61 also shows that the efficiency decreases as A_2/A_1 increases and that

the effect is greatest for large values of 2θ. At the same rate of expansion, that is, at the same increase in area per unit length of duct, the circular section is more efficient than the square section and the square section is more efficient than the rectangular section.

There is less tendency for separation to occur and the efficiency is better for a thin boundary layer at the diffuser entrance than for a thick one. The efficiency therefore is better for a short duct ahead of the diffuser than for a long one. This fact, however, is principally of academic interest and is not of much practical value in designing ducts for aircraft, for which duct length is generally dictated by other factors.

Because the static pressure continues to rise for some distance from the exit of a diffuser, the efficiency of a diffuser is greater when it is followed by a length of outlet duct. For $2\theta = 8^\circ$, the efficiency may be increased 5 to 15 percent by an outlet duct. The effect is greater when 2θ is larger than 8°. If the entrance velocity distribution is constant, the outlet duct length should be about $4D_2$; if the entrance velocity distribution is that of turbulent flow, the length should be about $6D_2$.

When a quick expansion must be used, separation can be delayed and the efficiency increased by curvature of the walls. The benefit of curvature is most pronounced in the range $15^\circ < 2\theta < 30^\circ$. A good type of curvature seems to be one that begins gradually and ends in a straight line, as shown in figure 62.

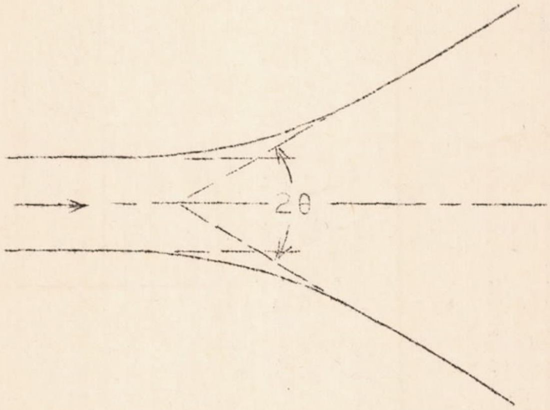

Figure 62. - Diffuser with curved walls.

Loss in total pressure at quite large angles of expansion may be reduced by use of deflectors. A system of deflectors, such as is shown in figure 63, can increase η by about 40 percent for $2\theta = 90^{\circ}$.

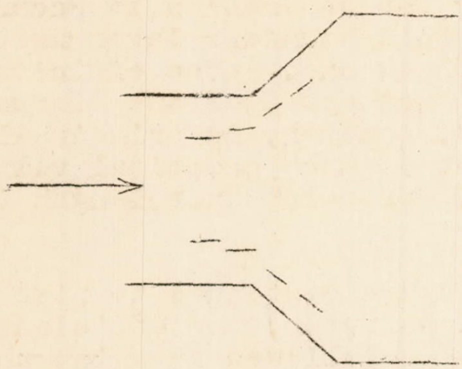

Figure 63. - Diffuser with deflectors.

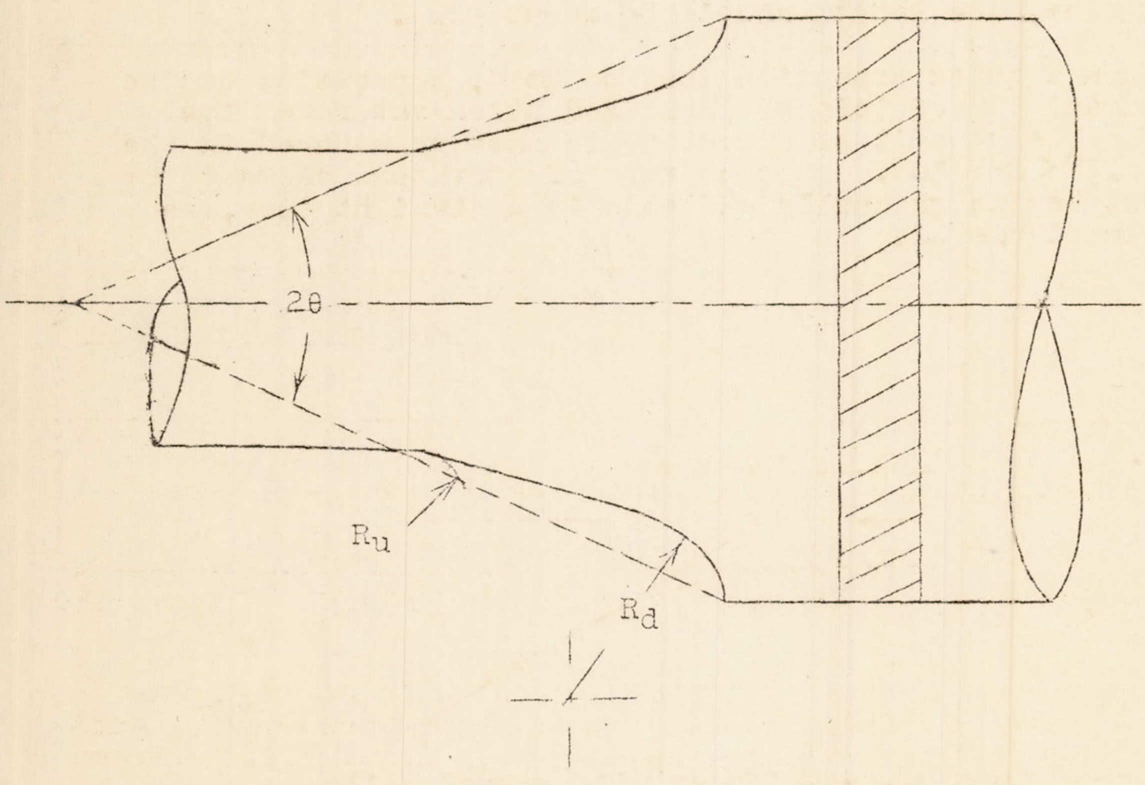

Figure 64. - Definition of diffuser symbols.

Diffusers Followed by Resistances

When a heat exchanger is directly behind a diffuser, the expansion can be made much more rapidly, for the same loss of total pressure, than when a large resistance is not behind the diffuser. Tests of diffuser-resistance combinations were reported in reference 28, from which the material in the present section is taken. In these tests, the diffuser shape was defined by three variables: a radius R_u at the entrance of the diffuser, an angle 2θ of a cone, and a radius R_d at the exit of the diffuser. (See fig. 64.)

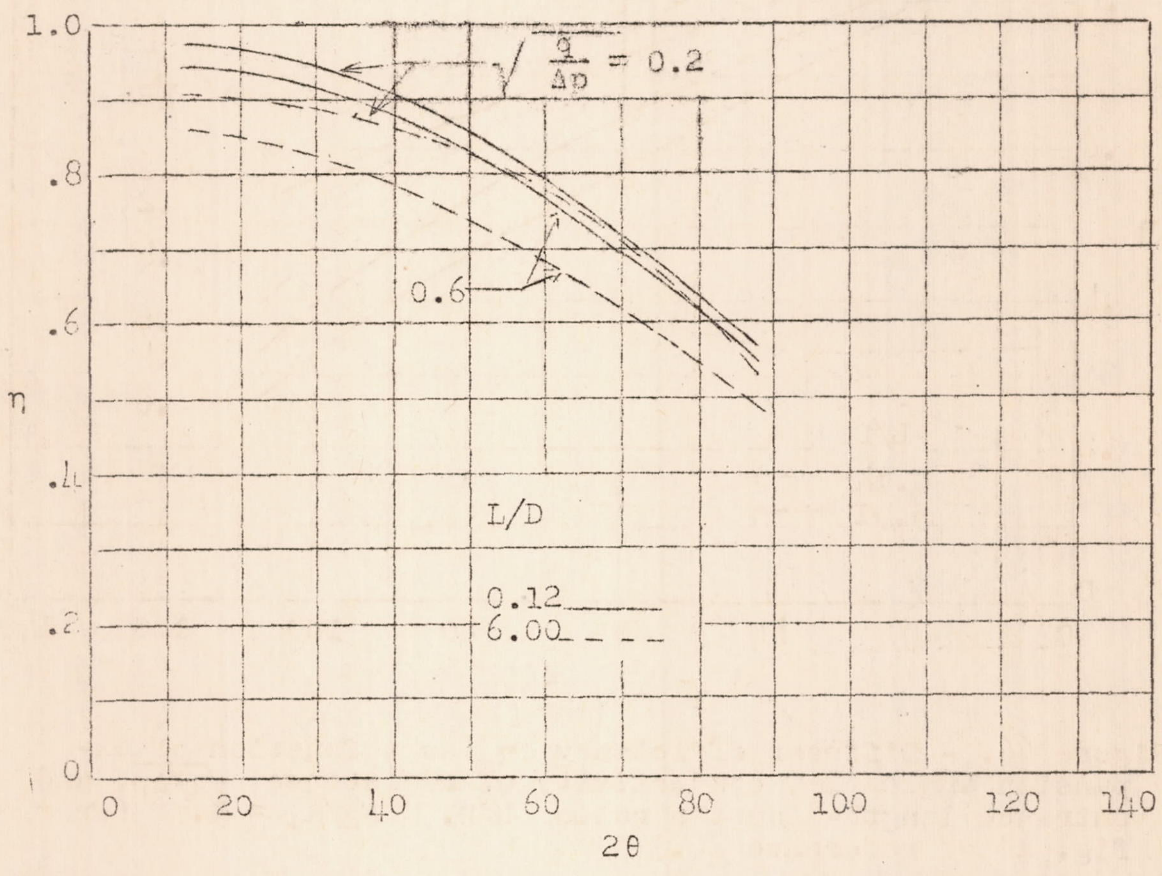

Figure 65. - Diffuser efficiency η as a function of expansion angle 2θ, conductivity of resistance $\sqrt{q/\Delta p}$, and entrance length-diameter ratio L/D. $A_2/A_1 = 2$. (From fig. 19 of reference 28.)

Figures 65 and 66 show the efficiency η of a diffuser that is followed by a resistance, as affected by the conductivity of the resistance $\sqrt{q/\Delta p}$, the ratio of expansion A_2/A_1, and the L/D of the entrance duct, that is, the velocity distribution at the diffuser entrance.

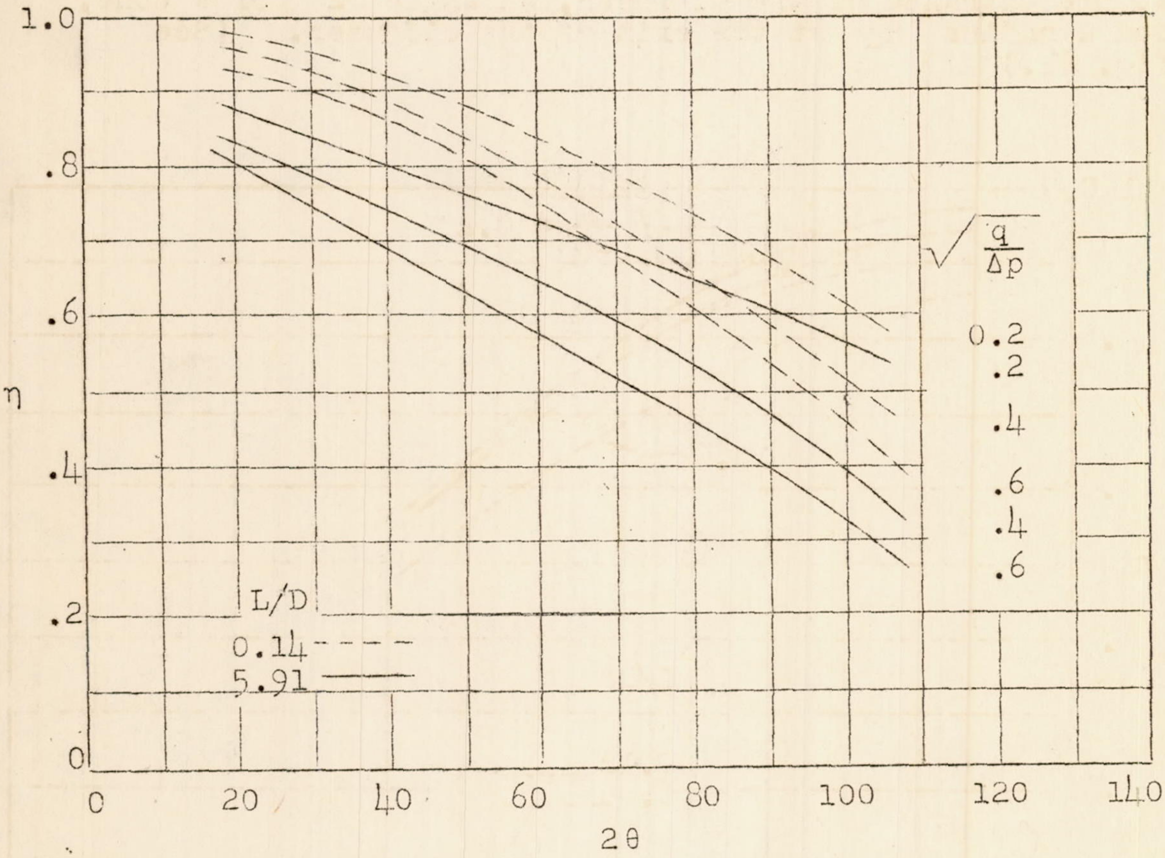

Figure 66. - Diffuser efficiency η as a function of expansion angle 2θ, conductivity of resistance $\sqrt{q/\Delta p}$, and entrance length-diameter ratio L/D. $A_2/A_1 = 3$. (From fig. 20 of reference 28.)

The tests showed that the value of R_u is not critical. It need be only large enough to remove the abruptness of the corner. A value of $R_d = D_2/3$ appeared better than other values tested, except when the diffuser was very short. When the diffuser was very short, the beginning of the curve was too abrupt and a value of $R_d = D_2/6$ was better. The effective angle of expansion 2θ is as shown in figure 64.

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APPENDIX A

PHYSICAL PROPERTIES OF AIR

The specific heat of air at constant pressure may be taken as

$$c_p = 0.238 \ \text{Btu/lb/}^{\circ}\text{F}$$

for all temperatures and pressures encountered in heat-exchanger work.

The Prandtl number for air can be considered a constant and equal to

$$\frac{c_p \mu g}{k} = 0.76$$

For air, the ratio of the specific heat at constant pressure to the specific heat at constant volume is

$$\gamma = 1.405$$

The coefficient of viscosity μ of air and the thermal conductivity k of air can be considered functions of temperature only. These quantities are shown as functions of temperature in $^{\circ}$F in figure 67. The value of μ at 68° F was taken from reference 29 and the value of k at 32° F was taken from reference 30. Values of both μ and k at other temperatures were calculated by Sutherland's equation given in reference 30.

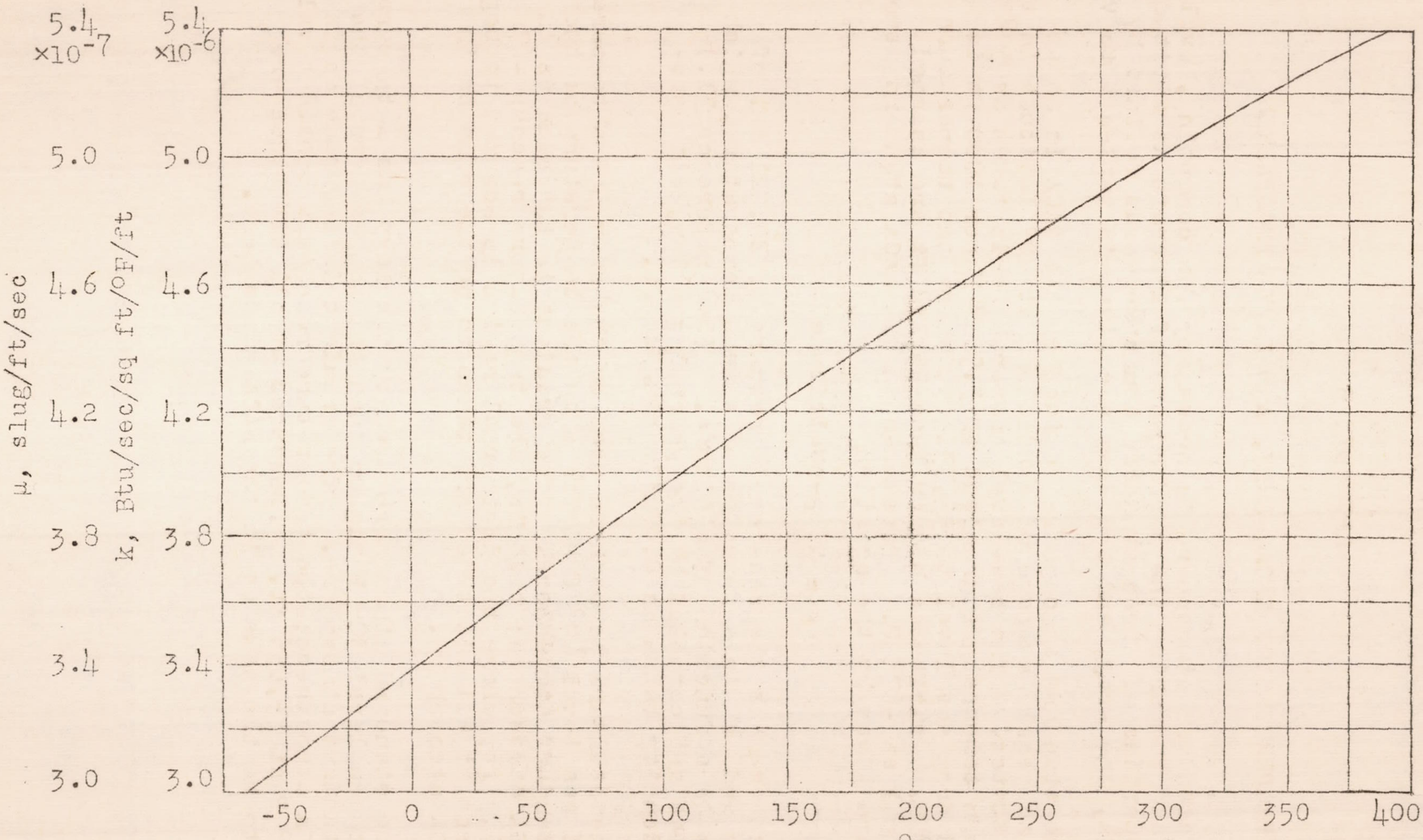

Figure 67. - Coefficient of viscosity μ and thermal conductivity k of air as functions of temperature.

APPENDIX B

PROPERTIES OF NACA, ARMY, AND NAVY ATMOSPHERES

A standard atmosphere is necessary for computing, evaluating, and comparing the performance of aircraft at altitude. The properties of three standard atmospheres as defined by the NACA, the Army, and the Navy are given in tables V to VII.

The standard atmosphere defined by the NACA is an approximate yearly average of conditions at 40° latitude in the United States. The sea-level temperature is taken as 59° F. A uniform temperature gradient of 3.566° F per 1000 ft is assumed from sea level to 35,332 ft, where the temperature is -67° F. Altitudes higher than 35,332 ft are assumed isothermal at -67° F. Pressures in the NACA standard atmosphere are given by the equation

$$p = p_0 \, e^{-c \times \text{altitude}}$$

in which p_0 is the sea-level pressure of 29.921 in. of mercury and c is a function of a mean temperature, which in turn is a function of altitude. (See reference 31.) Densities are computed by the general gas law, with sea-level density taken as 0.002378 slug/cu ft.

The standard atmosphere used by the Army has a temperature at sea level 40° F higher than the temperature of the NACA standard atmosphere. The gradient is the same as in the NACA atmosphere; that is, the temperature reaches -67° F at 46,549 ft, where the isothermal region is assumed to begin. Pressures at all altitudes are taken to be the same as in the NACA atmosphere.

The standard atmosphere used by the Navy has a temperature at sea level 30° F higher than the temperature of the NACA standard atmosphere. The gradient is the same as in the NACA atmosphere; the temperature reaches a constant value of -67° F at 43,656 ft. Pressures at all altitudes are taken to be the same as in the NACA atmosphere.

TABLE V

NACA STANDARD ATMOSPHERE

Altitude	Temperature (°F)	Pressure (in. Hg)	Density, ρ (slug/cu ft)	Relative density, ρ/ρ_o
0	59.0	29.92	0.002378	1.0000
1,000	55.4	28.86	.002309	.9710
2,000	51.9	27.82	.002242	.9428
3,000	48.3	26.81	.002176	.9151
4,000	44.7	25.84	.002112	.8881
5,000	41.2	24.89	.002049	.8616
6,000	37.6	23.98	.001988	.8358
7,000	34.0	23.09	.001928	.8106
8,000	30.5	22.22	.001869	.7859
9,000	26.9	21.38	.001812	.7619
10,000	23.3	20.58	.001756	.7384
11,000	19.8	19.79	.001702	.7154
12,000	16.2	19.03	.001648	.6931
13,000	12.6	18.29	.001596	.6712
14,000	9.1	17.57	.001545	.6499
15,000	5.5	16.88	.001496	.6291
16,000	1.9	16.21	.001448	.6088
17,000	-1.6	15.56	.001401	.5891
18,000	-5.2	14.94	.001355	.5698
19,000	-8.8	14.33	.001311	.5509
20,000	-12.3	13.75	.001267	.5327
21,000	-15.9	13.18	.001225	.5148
22,000	-19.5	12.63	.001183	.4974
23,000	-23.0	12.10	.001143	.4805
24,000	-26.6	11.59	.001103	.4640
25,000	-30.2	11.10	.001065	.4480
26,000	-33.7	10.62	.001028	.4323
27,000	-37.3	10.16	.000992	.4171
28,000	-40.9	9.720	.000957	.4023
29,000	-44.4	9.293	.000922	.3879
30,000	-48.0	8.880	.000889	.3740
31,000	-51.6	8.483	.000857	.3603
32,000	-55.1	8.101	.000826	.3472
33,000	-58.7	7.732	.000795	.3343
34,000	-62.2	7.377	.000765	.3218
35,000	-65.8	7.036	.000736	.3098
36,000	-67.0	6.708	.000704	.2962
37,000	-67.0	6.395	.000671	.2824
38,000	-67.0	6.096	.000640	.2692
39,000	-67.0	5.812	.000610	.2566

TABLE V - Continued

NACA STANDARD ATMOSPHERE - Continued

Altitude	Temperature (°F)	Pressure (in. Hg)	Density, ρ (slug/cu ft)	Relative density, ρ/ρ_o
40,000	-67.0	5.541	0.000582	0.2447
41,000	-67.0	5.283	.000554	.2332
42,000	-67.0	5.036	.000529	.2224
43,000	-67.0	4.802	.000504	.2120
44,000	-67.0	4.578	.000481	.2021
45,000	-67.0	4.364	.000459	.1926
46,000	-67.0	4.160	.000437	.1837
47,000	-67.0	3.966	.000417	.1751
48,000	-67.0	3.781	.000397	.1669
49,000	-67.0	3.604	.000379	.1591
50,000	-67.0	3.436	.000361	.1517
51,000	-67.0	3.276	.000344	.1447
52,000	-67.0	3.124	.000328	.1379
53,000	-67.0	2.978	.000313	.1316
54,000	-67.0	2.839	.000298	.1253
55,000	-67.0	2.707	.000284	.1194
56,000	-67.0	2.581	.000271	.1140
57,000	-67.0	2.460	.000258	.1085
58,000	-67.0	2.346	.000246	.1034
59,000	-67.0	2.236	.000235	.0988
60,000	-67.0	2.132	.000224	.0942
61,000	-67.0	2.033	.000214	.0900
62,000	-67.0	1.938	.000204	.0858
63,000	-67.0	1.847	.000194	.0816
64,000	-67.0	1.761	.000185	.0778
65,000	-67.0	1.679	.000176	.0740
66,000	-67.0	1.601	.000168	.0706
67,000	-67.0	1.526	.000160	.0673
68,000	-67.0	1.455	.000153	.0643
69,000	-67.0	1.387	.000146	.0614
70,000	-67.0	1.322	.000139	.0585
71,000	-67.0	1.261	.000132	.0555
72,000	-67.0	1.202	.000126	.0530
73,000	-67.0	1.146	.000120	.0505
74,000	-67.0	1.093	.000115	.0484
75,000	-67.0	1.041	.000109	.0458
76,000	-67.0	.993	.000104	.0437
77,000	-67.0	.946	.000099	.0416
78,000	-67.0	.902	.000095	.0399
79,000	-67.0	.860	.000090	.0378
80,000	-67.0	.820	.000086	.0362

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TABLE VI

ARMY SUMMER STANDARD ATMOSPHERE

Altitude	Temperature (°F)	Pressure (in. Hg)	Density, ρ (slug/cu ft)	Relative density, ρ/ρ_0
0	99.0	29.92	0.002207	1.0000
1,000	95.4	28.86	.002143	.9710
2,000	91.9	27.82	.002081	.9429
3,000	88.3	26.81	.002017	.9139
4,000	84.7	25.84	.001958	.8872
5,000	81.2	24.89	.001897	.8595
6,000	77.6	23.98	.001841	.8342
7,000	74.0	23.09	.001783	.8079
8,000	70.5	22.22	.001729	.7834
9,000	66.9	21.38	.001674	.7585
10,000	63.3	20.58	.001623	.7354
11,000	59.8	19.79	.001571	.7118
12,000	56.2	19.03	.001521	.6892
13,000	52.6	18.29	.001472	.6670
14,000	49.1	17.57	.001423	.6448
15,000	45.5	16.88	.001378	.6244
16,000	41.9	16.21	.001332	.6035
17,000	38.4	15.56	.001289	.5841
18,000	34.8	14.94	.001245	.5641
19,000	31.2	14.33	.001205	.5460
20,000	27.7	13.75	.001163	.5270
21,000	24.1	13.18	.001123	.5088
22,000	20.5	12.63	.001085	.4916
23,000	17.0	12.10	.001047	.4744
24,000	13.4	11.59	.001010	.4576
25,000	9.8	11.10	.000974	.4413
26,000	6.3	10.62	.000940	.4259
27,000	2.7	10.16	.000907	.4110
28,000	-.9	9.720	.000874	.3960
29,000	-4.4	9.293	.000841	.3811
30,000	-8.0	8.880	.000811	.3675
31,000	-11.6	8.483	.000781	.3539
32,000	-15.1	8.101	.000752	.3407
33,000	-18.7	7.732	.000723	.3276
34,000	-22.2	7.377	.000695	.3149
35,000	-25.8	7.036	.000668	.3027
36,000	-29.4	6.708	.000643	.2914
37,000	-32.9	6.395	.000618	.2800
38,000	-36.5	6.096	.000594	.2691
39,000	-40.1	5.812	.000571	.2587
40,000	-43.6	5.541	.000549	.2488
41,000	-47.2	5.283	.000528	.2392
42,000	-50.8	5.036	.000508	.2302
43,000	-54.3	4.802	.000488	.2211
44,000	-57.9	4.578	.000470	.2130
45,000	-61.5	4.364	.000452	.2048
46,000	-65.0	4.160	.000434	.1967
47,000	-67.0	3.966	.000417	.1889
48,000	-67.0	3.781	.000397	.1799
49,000	-67.0	3.604	.000379	.1717

TABLE VII

NAVY SUMMER STANDARD ATMOSPHERE

Altitude	Temperature (^{o}F)	Pressure (in. Hg)	Density, ρ (slug/cu ft)	Relative density, ρ/ρ_o
0	89.0	29.92	0.002248	1.0000
1,000	85.4	28.86	.002183	.9711
2,000	81.9	27.82	.002118	.9422
3,000	78.3	26.81	.002054	.9137
4,000	74.7	25.84	.001993	.8866
5,000	71.2	24.89	.001933	.8599
6,000	67.6	23.98	.001875	.8341
7,000	64.0	23.09	.001818	.8087
8,000	60.5	22.22	.001761	.7834
9,000	56.9	21.38	.001706	.7589
10,000	53.3	20.58	.001654	.7358
11,000	49.8	19.79	.001601	.7122
12,000	46.2	19.03	.001551	.6899
13,000	42.6	18.29	.001501	.6677
14,000	39.1	17.57	.001452	.6459
15,000	35.5	16.88	.001405	.6250
16,000	31.9	16.21	.001359	.6045
17,000	28.4	15.56	.001314	.5845
18,000	24.8	14.94	.001271	.5654
19,000	21.2	14.33	.001228	.5463
20,000	17.7	13.75	.001187	.5280
21,000	14.1	13.18	.001147	.5102
22,000	10.5	12.63	.001107	.4924
23,000	7.0	12.10	.001069	.4755
24,000	3.4	11.59	.001032	.4591
25,000	-.2	11.10	.000996	.4431
26,000	-3.7	10.62	.000960	.4270
27,000	-7.3	10.16	.000926	.4119
28,000	-10.9	9.720	.000893	.3972
29,000	-14.4	9.293	.000860	.3826
30,000	-18.0	8.880	.000829	.3688
31,000	-21.6	8.483	.000798	.3550
32,000	-25.1	8.101	.000768	.3416
33,000	-28.7	7.732	.000739	.3287
34,000	-32.2	7.377	.000711	.3163
35,000	-35.8	7.036	.000684	.3043
36,000	-39.4	6.708	.000658	.2927
37,000	-43.0	6.395	.000633	.2816
38,000	-46.6	6.096	.000608	.2705
39,000	-50.2	5.812	.000585	.2602
40,000	-53.8	5.541	.000561	.2496
41,000	-57.4	5.283	.000541	.2407
42,000	-61.0	5.036	.000521	.2318
43,000	-64.6	4.802	.000501	.2229
44,000	-67.0	4.578	.000481	.2140

APPENDIX C

IMPACT-PRESSURE CHART

Impact pressure is shown in figure 68 as a function of speed and altitude. Impact pressure is the pressure shown by a pitot-static tube and is defined as the total pressure minus the static pressure, that is, the static pressure at a stagnation point minus the static pressure in the free stream.

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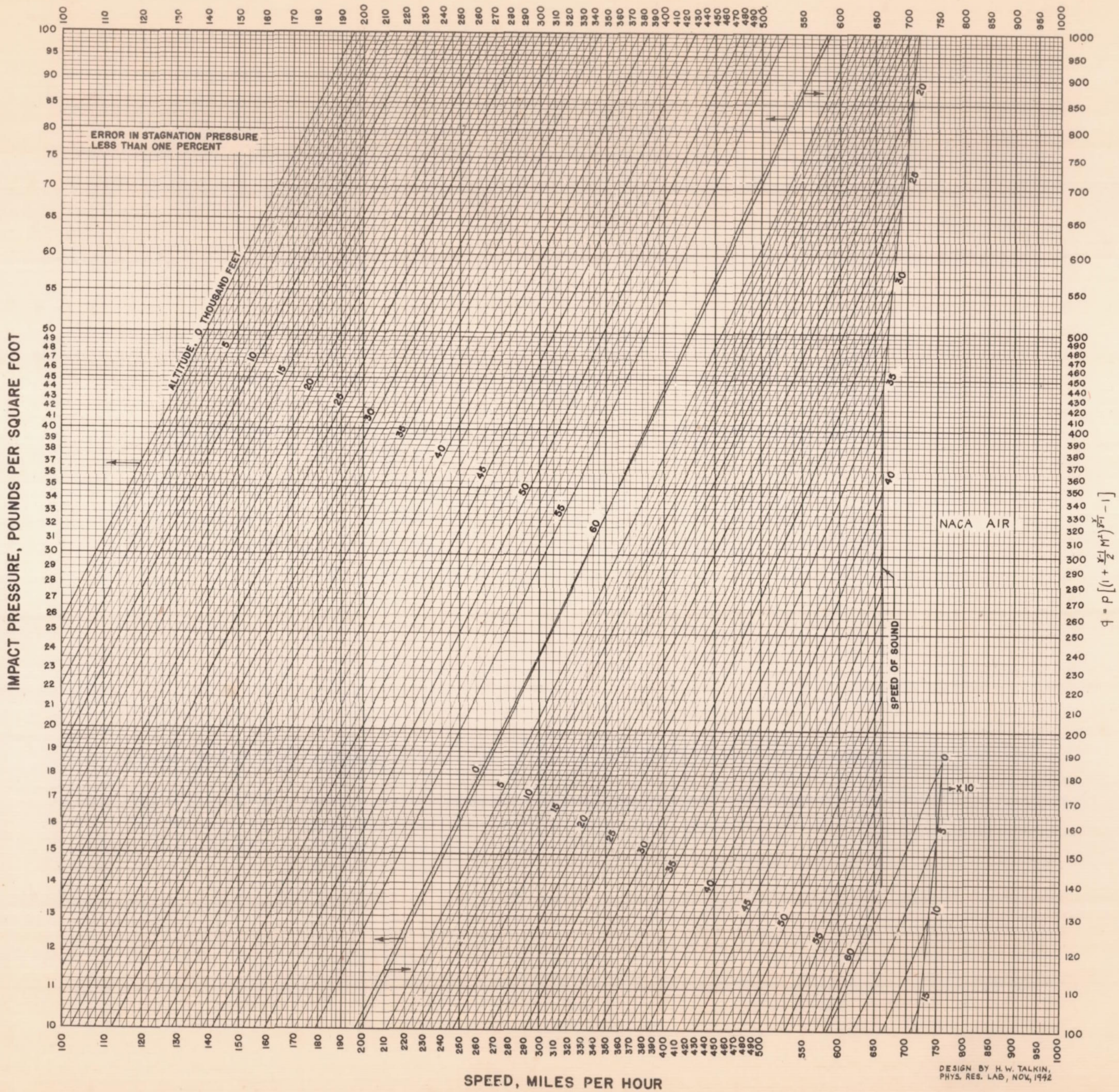

Figure 68. - Impact pressure as a function of speed and altitude.